STEAM BOILERS

AND BOILER ACCESSORIES

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FOR STEAM USERS, ENGINEERS, AND ENGINEERING STUDENTS

BY

W. INCHLEY, B.Sc., A.M.I.Mech.E.

LECTURER ON ENGINEERING AT UNIVERSITY COLLEGE, NOTTINGHAM

ILLUSTRATED

NEW YORK

LONGMANS, GREEN, AND CO.

London: EDWARD ARNOLD

1912

PREFACE

In this work an attempt has been made to produce a book of reasonable size which deals with the construction and working of Steam Boilers and their Accessories in a manner suitable for the requirements of steam users and engineering students generally. Technical details of boiler construction, such as the pitch of rivets, strength of plates, etc., are designedly omitted, since few readers will find themselves called upon to have expert knowledge on such questions, which must be left to the responsible boiler manufacturer. Only general principles of the strength of parts are here included. Chapter I. briefly explains the various types of boilers in common use together with the materials used in their construction, after which the theory of combustion and steam generation is considered in Chapters II.–V.

The method of treatment has been to avoid as far as possible the use of advanced mathematics—a difficult task when writing on the subject of Heat Transmission. In Chapter V. prominence has been given to the effect on boiler performance of using high gas speeds, and the Author is indebted to Professor J. T. Nicolson, D.Sc., for much of the matter included therein. A detailed description of the various types of Smoke-Tube Boilers is given in Chapter VI., whilst Chapter VII. is devoted to Water-Tube Boilers. In a book of this scope it has been found impossible to describe *all* the different types, but it is hoped that the majority of those boilers which have proved satisfactory in practice have been included.

The various Boiler Accessories and Mountings representative of modern practice have been treated in Chapters VIII.

and IX. Chapter X. deals exclusively with the selection and treatment of Boiler Feed Water, Chapter XI. with the practical working of Boilers, and Chapter XII. with Boiler Trials.

Free use has been made of the technical journals and the Proceedings of the various Engineering Institutions, information from which is acknowledged as far as possible in the text. In particular, the Author's thanks are due to Mr. C. E. Stromeyer for permission to reproduce portions of his memorandum to the Manchester Steam Users' Association for the year 1909; to the Council of the Institution of Civil Engineers; the Council of the Institution of Mechanical Engineers; the American Society of Mechanical Engineers; and the Engineers' Society of Western Pennsylvania.

Acknowledgment and thanks are also due to the numerous firms who have supplied the Author with information and drawings from which many of the illustrations have been prepared, particularly to Messrs. Babcock & Wilcox, Ltd.; Messrs. Clarke, Chapman & Co., Ltd.; Messrs. Galloways, Ltd.; The Stirling Boiler Co., Ltd.; Messrs. Schäffer & Budenberg, Ltd.; Messrs. J. Hopkinson & Co., Ltd.; Messrs. Marshall, Sons, & Co., Ltd.; The British Niclausse Boiler Co., Ltd.; Messrs. John I. Thornycroft & Co., Ltd.; The Cambridge Scientific Instrument Co., Ltd.; Messrs. Yarrow & Co., Ltd.; Messrs. Alley & MacLellan, Ltd.; and Messrs. Cochran & Co., Ltd., as well as others who have also kindly assisted in this respect.

W. INCHLEY

University College,
Nottingham,
August, 1912

CONTENTS

CHAPTER I

TYPES OF BOILERS AND THE MATERIALS USED IN THEIR CONSTRUCTION

CHAPTER II

FUELS AND COMBUSTION

CHAPTER III

BOILER FURNACES

CHAPTER IV

DRAUGHT

CHAPTER V

GENERATION OF STEAM AND HEAT TRANSMISSION

CHAPTER VI

CYLINDRICAL BOILERS OF THE SMOKE-TUBE TYPE

CHAPTER VII

WATER-TUBE BOILERS

CHAPTER VIII

BOILER ACCESSORIES

CHAPTER IX

BOILER MOUNTINGS AND STEAM PIPES

CHAPTER X

THE SELECTION AND TREATMENT OF A BOILER FEED WATER

CHAPTER XI

THE PRACTICAL RUNNING OF BOILERS

CHAPTER XII

STEAM BOILER TRIALS

LIST OF FOLDING PLATES

STEAM BOILERS

CHAPTER I

TYPES OF BOILERS AND THE MATERIALS USED IN THEIR CONSTRUCTION

1. Function of a Steam Plant.—The function of a steam plant is to convert, with the greatest economy, as much as possible of the heat energy contained in the fuel into a form in which it can be usefully turned to account in practice. This cannot be accomplished without losing a very large proportion of the original heat in the fuel. It will be instructive if we briefly consider what proportion of the heat in the fuel may be actually available for useful purposes. Taking for our illustration an electrical generating plant using coal which has a calorific value of 14,000 British Thermal Units per pound (in other words, 1 pound of the coal contains about 14,000 × 778 = 11,000,000 foot-pounds of energy), we may trace the various transformations of energy which take place from 1 pound of coal as follows:—

			Foot-pounds.
Heat energy in 1 pound of coal		=	11,000,000
Heat energy in the boiler steam	= 0·7 × 11,000,000	=	7,700,000
Heat energy converted into mechanical work in the engine cylinder	= 0·1 × 7,700,000	=	770,000
Energy available on the engine crankshaft	= 0·85 × 770,000	=	654,500
Electrical energy leaving dynamo	= 0·90 × 654,500	=	589,000
Energy appearing say as light from the electric lamps	= 0·015 × 589,000	=	8,800

The above rough estimate neglects losses between the boiler and engine due to condensation in the steam pipes, and also neglects the electrical losses in the cables from the dynamo to the lamps. The various efficiencies assumed above are average values which might be obtained in actual practice. With the losses which occur between the boiler and the lamps we are not concerned in this book, our purpose being to consider the boiler only.

The function of a boiler then is to generate steam to be used in engine cylinders for doing useful work. That boiler is the most economical in working which will transfer into the steam the greatest amount of the heat contained in a certain weight of fuel.

2. Types of Boilers.—There are a variety of different types of boilers used in practice, all of which may be broadly classified under two types, namely, *Smoke-Tube Boilers* and *Water-Tube Boilers*. Smoke-tube, or multitubular boilers are those in which the products of combustion of the fuel pass along *inside* the tubes, the tubes themselves being surrounded by water. Water-tube, or tubulous boilers are those in which the water and steam pass along inside the tubes, the tubes themselves being surrounded by the products of combustion.

The boiler furnace may be arranged in one of two ways. In *Internally Fired* boilers the furnace forms part of the boiler itself, being almost, if not quite, surrounded by water. Boilers of the smoke-tube type are usually internally fired. In *Externally Fired* boilers, the furnace is exterior to the boiler proper—the boiler being considered as the steam-generating portion—as is the case with all water-tube boilers, and some smoke-tube boilers, notably the externally fired multitubular boiler and the Lancashire boiler when arranged for burning refuse fuels.

In any type of boiler, whether smoke-tube, water-tube, internally or externally fired, the actual area of the furnace fire-grate is called the *Grate Area;* the superficial area of the plates and tubes *on the gas side* which are in contact with the water on one side, and with the hot gases on the other side, is called the *Heating Surface* of the boiler.

There are two fundamental principles underlying the

working of all types of steam boilers, namely, that only a certain quantity of fuel can be economically burned per hour per square foot of grate area under given working conditions, and that only a certain quantity of water can be evaporated per pound of fuel under the same conditions of working. The quantity of heat which passes through the heating surface of the boiler is all that is available for generating steam, and if the heating surface is too small, the heat will not be able to pass so readily through it and therefore more will be lost up the chimney; on the other hand, if the heating surface is too large, the extra cost of that surface together with the size of the boiler may more than counterbalance the gain due to less heat being carried away by the gases. There is thus a certain *ratio of heating surface to grate area* which will give the most economical results in a financial sense. The value of this ratio depends upon the particular type of boiler under consideration, *i.e.* upon whether the boiler is of the smoke-tube type, internally or externally fired, or is of the water-tube type. In any case the best ratio is a matter of practical experience and experiment. A brief comparison of the chief types of boilers with their leading dimensions is given in Table I., in which, unless other wise stated, all boilers are arranged for hand firing (pp. 4–11).

3. Choice of a Boiler.—The best type of boiler to adopt depends upon certain circumstances which will require the careful consideration of the steam user. It is impossible to lay down a general rule to decide on the choice to be made, as it depends entirely upon the conditions under which the boiler will have to work. The chief factors to be considered may be briefly summed up as follows:—

1. Steam pressure required.
2. Rate of evaporation required, *i.e.* the number of pounds of steam required per hour.
3. Nature of the load, whether steady or fluctuating.
4. Floor space and head room available.

Speaking generally, when the necessary floor space is available, the Lancashire boiler has proved itself a reliable and efficient steam generator, particularly when the load is steady and the steam pressure not too high. When the floor space is

TABLE I.

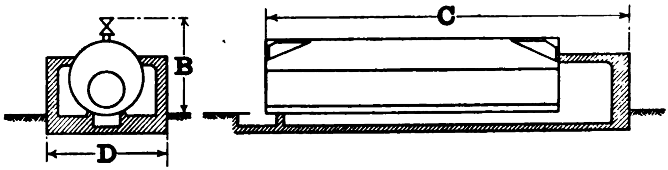


FIG. 1.—CORNISH BOILER. Shell, 6 ft. dia., 24 ft. long; one internal furnace tube, 3 ft. dia.

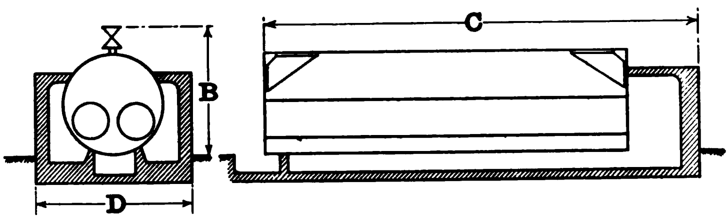


FIG. 2.—LANCASHIRE BOILER. Shell, 8 ft. 6 in. dia., 30 ft. long; two internal furnace tubes, each 3 ft. 6 in. dia.

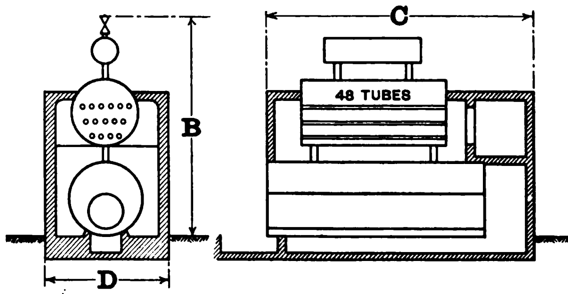


FIG. 3.—COMBINED CORNISH AND MULTITUBULAR BOILER. Cornish portion, 6 ft. dia., 18 ft. long; one furnace tube, 3 ft. 6 in dia.; multitubular, 5 ft. 6 in. dia., 12 ft. long, with 48 tubes, each 4 in. dia.; steam dome, 2 ft. dia., 8 ft. long.

Type of boiler.	Pounds of water evaporated per hour from and at 212° F.	Heating surface in sq. ft.	Grate area in sq. ft.	Ratio $\frac{\text{Heating surface}}{\text{Grate area}}$	Pounds of water per hour per sq. ft. of heating surface.	Pounds of water per hour per sq. ft. of grate area.	Dimensions over brickwork— B, Height to stop valve. C, Length from front to back. D, Width.
Fig. 1.—Cornish . . .	2600	492	18	27·2	5·3	144	B 9′ 6″ C 30′ 0″ D 9′ 11″
Fig. 2.—Lancashire . . .	7800	1150	42	27·2	6·8	186	B 11′ 6″ C 36′ 6″ D 12′ 10″
Fig. 3.—Combined Cornish and multitubular . .	2600	1370	19	72	1·9	137	B 20′ 0″ C 21′ 6″ D 10′ 0″

TABLE I.—*continued.*

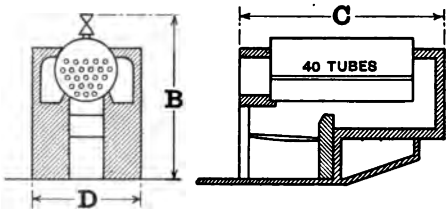


FIG. 4.—EXTERNALLY FIRED MULTITUBULAR BOILER. Shell, 5 ft. dia., 12 ft. long, with 40 tubes, each 4 in. dia.

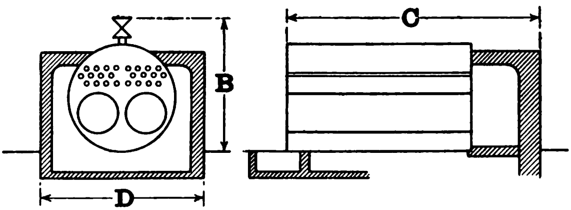


FIG. 5.—DRYBACK BOILER. Shell, 9 ft. dia., 15 ft. long, with 2 furnace tubes, each 3 ft. 3 in. dia.

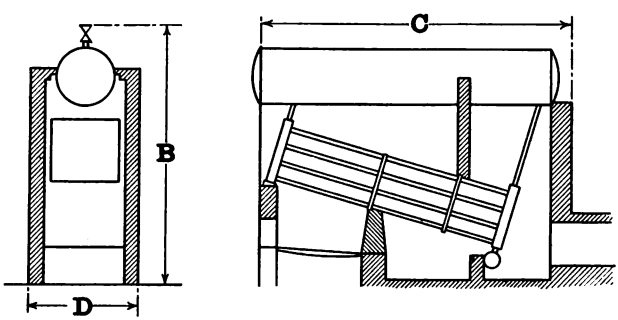


FIG. 6.—BABCOCK AND WILCOX WATER-TUBE BOILER. (Land type.)

Type of boiler.	Pounds of water evaporated per hour from and at 212° F.	Heating surface in sq. ft.	Grate area in sq. ft.	Ratio $\frac{\text{Heating surface}}{\text{Grate area}}$	Pounds of water per hour per sq. ft. of heating surface.	Pounds of water per hour per sq. ft. of grate area.	Dimensions over brickwork—B, Height to stop valve. C, Length from front to back. D, Width.
Fig. 4.—Externally fixed multitubular	2700	617	24	25·7	4·4	112	B 12′ 6″ C 17′ 6″ D 9′ 0″
Fig. 5.—Dry-back . . .	7800	1440	42	34·3	5·4	186	B 15′ 3″ C 21′ 0″ D 13′ 6″
Fig. 6.—Babcock and Wilcox . .	8,000	2255	43*	52·3	3·5	186	B 18′ 9⅜″ C 23′ 0″ D 9′ 2″
	12,000	3240	58*	55·8	3·7	206	B 18′ 2½″ C 23′ 0″ D 12′ 8″
	16,000	4020	76*	53	3·9	210	B 19′ 3″ C 23′ 0″ D 13′ 10″
	30,000	8283	168 †	49·3	3·6	178	B 23′ 10⅝″ C 23′ 6″ D 19′ 1″

* Hand fired. If mechanically fired, add 12″ to height B.

† Mechanically fired. The largest size made by Messrs. Babcock and Wilcox, and always fitted with mechanical stokers.

TABLE I.—*continued.*

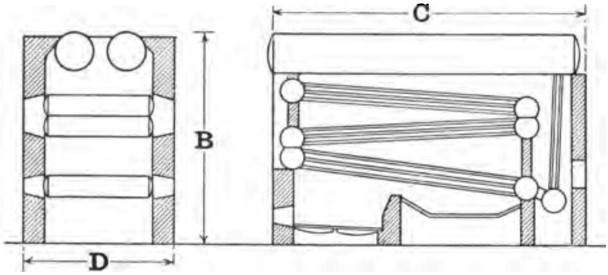


FIG. 7.—VOGT WATER-TUBE BOILER.

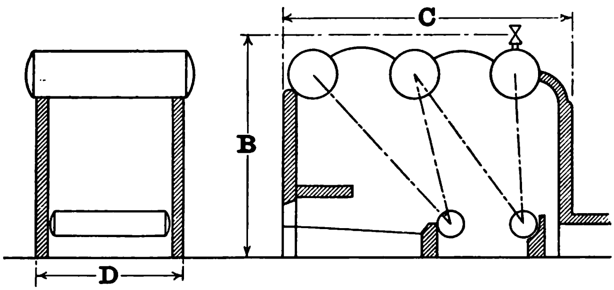


FIG. 8.—STIRLING BOILER (Land type).

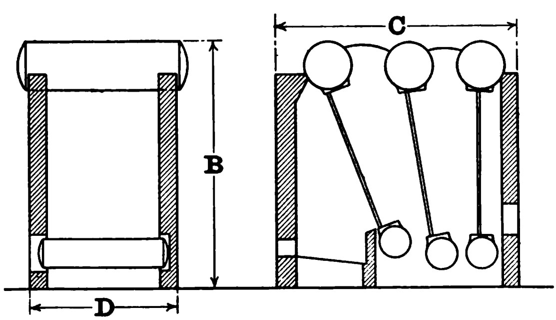


FIG. 9.—CLARKE CHAPMAN OR "WOODESON" BOILER.

Type of boiler.	Pounds of water evaporated per hour from and at 212° F.	Heating surface in sq. ft.	Grate area in sq. ft.	Ratio $\frac{\text{Heating surface}}{\text{Grate area}}$	Pounds of water per hour per sq. ft. of heating surface.	Pounds of water per hour per sq. ft. of grate area.	Dimensions over brickwork— B, Height to stop valve. C, Length from front to back. D, Width.
Fig. 7.—Vogt .	12,000	4000	76	52·7	3·0	158	B 19′ 9″ C 24′ 7″ D 12′ 5″
Fig. 8.—Stirling . . .	8,130	2197	45·5	48·4	3·7	179	B 18′ 0″ C 22′ 0″ D 9′ 6″
	12,000	3234	65	49·7	3·7	184	B 21′ 0″ C 22′ 3″ D 10′ 6″
	19,200	5183	90	57·6	3·7	212	B 21′ 0″ C 22′ 3″ D 14′ 6″
	39,500 *	10,675	204	52·3	3·7	195	B 24′ 0″ C 23′ 0″ D 24′ 0″
Fig. 9.—Clarke Chapman .	7,360	2025	32	63·4	3·6	230	B 22′ 9″ C 21′ 0″ D 7′ 10″
	12,000	3450	52	66·4	3·5	230	B 22′ 9″ C 21′ 0″ D 11′ 0″
	19,090	5400	83	65·0	3·5	230	B 22′ 9″ C 21′ 0″ D 15′ 9″
	25,600 †	5400	90	60·0	4·7	284	B 22′ 9″ C 21′ 0″ D 15′ 9″

* Largest size made for hand firing.
† Mechanically fired.

TABLE I.—*continued.*

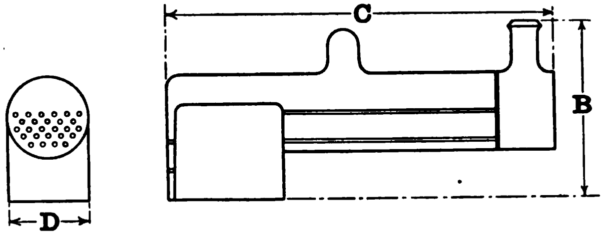


FIG. 10.—LOCOMOTIVE BOILER.

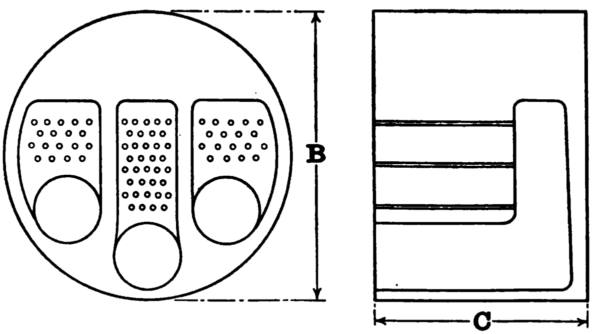


FIG. 11.—SCOTCH MARINE BOILER.

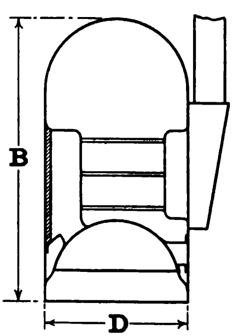


FIG. 12.—COCHRAN VERTICAL MULTITUBULAR BOILER. 7 ft. dia., 143 tubes each $2\frac{1}{2}$ in. dia.; funnel, 25 in. dia., 32 ft. high.

Type of boiler.	Pounds of water evaporated per hour from and at 212° F.	Heating surface in sq. ft.	Grate area in sq. ft.	Ratio $\frac{\text{Heating surface}}{\text{Grate area}}$	Pounds of water per hour per sq. ft. of heating surface.	Pounds of water per hour per sq. ft. of grate area.	Dimensions over brickwork—B, Height to stop valve. C, Length from front to back. D, Width.
Fig. 10.—216 tubes . . .	—	1456	28·4	51·2	—	—	B 9′ 6″ C 24′ 0″ D 4′ 8″
444 tubes . .	—	3670	57	64·3	—	—	B 9′ 9″ C 30′ 10″ D 6′ 10″
Fig. 11.—Scotch marine, 316 tubes . . .	15,000	2033	33	61·5	7·4	455	B 15′ C 10′ D 15′
Fig. 12.—Cochran vertical, 143 tubes . .	2750	500	26·75	18·7	5·5	103	B 14′ C 7′ D 7′

restricted and the necessary head room is available, water-tube boilers are more satisfactory, particularly when rapid steam raising and a high steam pressure are required. The advantages and disadvantages of the different types of boilers are discussed in detail in Chapters VI. and VII.

4. Materials used in Construction.—Cast Iron.—Cast iron should never be used in boiler construction in any place where it will be subjected to tension. Its ultimate strength in tension is usually from 7 to 10 tons per square inch; in compression, however, it is often about 50 tons per square inch. It is very brittle and of an uncertain character; owing to the liability to porosity, initial stress in cooling, etc., the working stress allowable in cast iron should not exceed about 1 ton per square inch in tension and 8 tons per square inch in compression. For the above reasons cast iron is avoided in all high-class boiler work, and when used at all, it is confined to seating blocks for boiler mountings and fittings and the like.

5. Malleable Cast Iron.—Although of a more reliable nature than cast iron, malleable cast iron is avoided in the best practice. Malleable castings are made by heating iron castings for long periods with red hematite in air-tight pots or boxes. The hematite removes some of the carbon from the cast iron and renders the casting stronger and much less brittle. It has about $1\frac{1}{2}$ times the strength of cast iron in tension. If employed on account of its cheapness instead of wrought iron or steel, its use should be confined to valve chests, safety-valve levers, and seatings on a boiler shell on which fittings are mounted.

6. Wrought Iron.—Wrought iron is a very ductile metal containing over 99 per cent. of pure iron, and only about one-tenth per cent. of carbon; its composition varies however with different qualities. It is desirable to keep phosphorus below 0·25 per cent., and sulphur below 0·05 per cent. Phosphorus makes the metal brittle when cold (cold short) and sulphur causes brittleness at a red heat (hot short). It comes from the puddling furnace in a spongy or pasty state and subsequent hammering and rolling do not expel all traces of slag, which may be traced in layers in the finished product. According to the amount

of rolling and working generally which wrought iron has received in the course of manufacture, and which improve its tenacity, it is known commercially as "Best," "Double Best," or "Treble Best." In the early days, wrought iron was the only material of which boiler shells were constructed, but in modern practice it has been replaced by mild steel, which is both stronger and cheaper.

Both the tenacity and ductility are greater with the grain than across it. Wrought-iron boiler plates should have an average ultimate strength in tension of 21 tons per square inch with the grain and of 19 tons per square inch across the grain, with an elongation with the grain of about 12 per cent. in a length of 10 inches.

In modern boiler construction its use is limited to those parts where welding is required, such as stay bars, tubes with swelled ends, and also rivets. A wrought-iron stay bar of good material should have an ultimate strength of about 23 tons per square inch with an elongation of not less than 20 per cent. in a length of 10 inches.

7. Mild Steel.—Mild steel is now the best and most usual material for boiler plates; it is practically homogeneous and possesses no structure similar to that of wrought iron. It is more ductile than wrought iron, as well as stronger and cheaper. For these reasons mild steel has almost entirely replaced wrought iron in boiler construction. It requires, however, more care in working than wrought iron and should be worked either at a red heat or quite cold, and not at a blue heat.

In welding steel it is important that the pieces to be united should contain the same amounts of carbon. If they do not, their welding temperatures are different. All straining of the material when cold by hammering or punching injures steel more than wrought iron, but the damage is entirely removed by annealing. Another great advantage of steel over wrought iron in construction is that plates of much greater area and weight can be obtained; steel plates of 70 square feet in area are obtained with little difficulty.

Board of Trade Regulations.—The following are the test requirements of mild steel plates for boiler construction.

Tensile Test.—Furnace plates and all plates exposed to flame, 26 to 28 tons per square inch with 20 per cent. elongation in 10 inches. Shell plates and plates not exposed to flame, 27 to 32 tons per square inch with 20 per cent. elongation in 10 inches. A test-piece must be cut from every plate.

Bending Test.—From every plate exposed to flame a strip is cut off, heated to a cherry red heat and cooled in water at 82° F. It must then stand bending to a curve, the inner radius of which is equal to one and a half times the thickness of the plate. In the case of shell plates and plates not exposed to flame, a test piece is taken from each plate and bent cold to a curve, the inner radius of which is equal to one and a half times the thickness of the plate.

Mild steel rivets are also frequently used in modern construction, and the bars from which they are made should have a tensile strength of from 26 to 30 tons per square inch with an elongation of 25 to 30 per cent. in 10 inches.

The strength and ductility of mild steel are largely influenced by the percentage of carbon which it contains. Boiler plates should not contain more than about 0·25 per cent. and rivet steel about 0·1 per cent.; the percentages of sulphur and phosphorus should be kept below that given for wrought iron in Art. 6.

8. Brittleness of Steel.—Several mysterious failures of mild steel have occurred from time to time without any apparent reason, and the possible liability of steel to become brittle with age has resulted in several papers being read before various Institutions. The conclusions arrived at by the Chief Engineer of the Manchester Steam Users' Association (Mr. C. E. Stromeyer) are embodied in a Memorandum to the Association for the year 1909, and from their suggestiveness and importance are here reproduced, by kind permission, at some length.

"Seeing that the mysterious failures could not be attributed to ageing, and as the other mechanical tests and the usual chemical analyses also failed to distinguish between those steels which had, and those steels which had not, failed in practice, it was decided to analyse for nitrogen. Our nitrogen determinations revealed at once that most of the mysterious failures were due either to this impurity or to the combined effects of

phosphorus and nitrogen as will be evidenced from the Table on p. 16. The chemical constituents—carbon, nitrogen and phosphorus—are given as well as the sums of the phosphorus and five times the nitrogen, these sums being, so it seems, a fair index of the brittleness of the material.

"It will be noticed that all those steels in which the sum of one phosphorus and five nitrogen exceeds 0·08 per cent. are brittle, or presumably so, while those in which the sum is under 0·08 are ductile, the only exceptions being sample T, which failed in the boiler shop, although the sum of P + 5N was only 0·049. In this case, however, no details could be obtained, the marine boiler works which sent it having been closed years ago, and the correspondence having been lost, but as the sample was flat it cannot have been part of a shell plate, and may have been the cracked part of an end plate, which plates, it is well known, do crack after flanging unless they are annealed. This sample, therefore, has no bearing on the matter, and might have been omitted. The same may be said of sample N, which failed, although the sum of P + 5N is also under 0·08, being 0·065. This sample formed part of a boiler shell in which caustic liquors were concentrated, which has the effect of imparting extreme brittleness to mild steel if it is in tension. The Table also suggests that the distinguishing feature between Bessemer and Open-Hearth Steel is that the former has a much larger percentage of nitrogen than the latter. . . .

"The most important conclusion to be drawn from this investigation, as far as it has progressed, is evidently that no enquiry into a failure of a steel plate can be considered complete unless the percentages of both phosphorus and nitrogen have been determined."

A number of other tests were also applied, such as the bending of samples with both sheared and planed edges, and also of nicked samples, but one and all failed to differentiate between reliable and unreliable steels, so that they were practically valueless.

In addition the following temper tests were carried out: "Three sets of samples were heated till they were cherry-red and then plunged into water at 82° F. Of these one set was

TABLE II.

Mark.		Carbon per cent.	Nitrogen per cent.	Phosphorus per cent.	5N+1P. per cent.
	The following steels were found to contain, relatively, much NITROGEN :—				
CD_2	Ship plates of basic Bessemer steel—failed in shipbuilding yard	0·090	0·023	0·062	0·177
Q_1	Basic steel. Boiler in Austrian boiler shop—burst under hydraulic test	0·127	0·020	0·052	0·152
PO_4	Furnace flange of 19-year old boiler—broke during repairs	—	0·018	—	—
L_1	German basic Bessemer steel. Bad quality supplied for the purpose of comparison	0·090	0·0153	0·060	0·136
B_4	Angle irons for small boiler. Broke in boiler shop. Almost certainly basic Bessemer steel	—	0·0146	—	—
H_1	German acid Bessemer steel, known to be unsuitable for boiler plates	0·210	0·0145	0·079	0·151
LO_5	Shell plate of old boiler. Cracked while in use	0·136	0·0127	0·147	0·210
CC_2	English acid Bessemer steel. Marine boiler shell burst under hydraulic test	0·130	0·0123	0·047	0·108
GT_4	German basic Bessemer steel. Gas vessel exploded at Innsbrück	—	0·0120	0·077	0·137
M_1	Butt strap of a water pipe which burst while in use. Believed to be basic Bessemer steel	0·090	0·0090	0·052	0·097
U_1	Russian boiler burst under hydraulic test when 6 years old. German or Russian steel	0·110	0·0066	0·052	0·085
	The following steels were found to contain, relatively, much PHOSPHORUS :—				
F_1	German basic open-hearth steel supplied for comparison	0·097	0·0043	0·210	0·231
R_1	British acid ” ” boiler shell plate cracked in bending rolls	0·165	0·0030	0·177	0·192
K_1	German basic ” ” found to contain too much P. Supplied for comparison	0·205	0·0041	0·095	0·115
E_1	German basic ” ” ” ” ” ” ”	0·165	0·0029	0·076	0·090
S_1	British acid ” ” shell plate edges broke while being sheared	0·135	0·0038	0·065	0·084
	The following steels contain only small quantities of NITROGEN *and* PHOSPHORUS :—				
C_1	German basic open-hearth steel. Hard quality	0·350	0·0033	0·054	0·070
Y_1	British acid ” ” High tenacity quality	0·275	0·0024	0·037	0·049
W_1	British acid ” ” Hard quality	0·270	0·0033	0·044	0·060
G_1	German acid ” ” Ordinary quality	0·215	0·0050	0·051	0·076
V_1	British acid ” ” ” ”	0·210	0·0035	0·045	0·062
B_1	German basic ” ” ” ”	0·200	0·0040	0·032	0·052
T_1	Fractured part of a thick flat plate from marine works. No details obtainable	0·200	0·0023	0·038	0·049
BB_1	British acid open-hearth steel. Ordinary quality	0·200	0·0032	0·029	0·045
J_1	German basic ” ” ” ” Supplied with K and L	0·195	0·0046	0·050	0·073
Z_1	British basic ” ” ” ”	0·195	0·0033	0·039	0·055
N_1	British acid ” ” Caustic soda boiler—shell plates cracked	0·185	0·0023	0·054	0·065
D_1	German basic ” ” Soft quality, but very rich in sulphur	0·127	0·0028	0·042	0·056
X_1	British basic ” ” Dead soft quality	0·122	0·0040	0·033	0·053
A_1	German basic ” ” ” ”	0·100	0·0031	0·026	0·041

bent till each sample broke; another set was bent down and up over an anvil edge till each sample broke, the number of bends being counted. The third set was treated in the same way, but after waiting twelve weeks. These results confirmed what has already been known, that these so-called temper tests are fairly reliable indicators of the amount of carbon present in the several steels. Thus C with 0·35 per cent. carbon broke off short on being bent. Y and W, both with 0·27 per cent. carbon, were fairly brittle under the alternate bending test, though W was markedly superior under the simple bending test. B, G, H, J, K, T, V, Z, BB were all nearly equal as regards percentage of carbon (0·195 to 0·210 per cent.), and their behaviours under the test were very similar, both H with much nitrogen and K with much phosphorus being slightly more ductile than the average. . . .

"Seeing that brittleness and unreliability in practice are largely due to phosphorus and nitrogen, which elements cannot be detected by the temper bending test, the question presents itself whether this test, which is now universally accepted as being able to discriminate between good and bad steel, is not standing in the way of progress towards the adoption of steels with higher tenacities than are now acceptable. Thus C, although its tenacity is only 33 tons, is, under present conditions, unacceptable, because it would not stand the temper tests, whereas Y and W with tenacities of 37·4 and 31·8 tons respectively would be accepted because their high percentages of silicon and manganese improve the testing qualities. Even better results for testing purposes would be obtained with steels rich in nitrogen, like Q, L, and H, which impurity adds 6·00, 4·35, and 4·59 tons respectively to the tenacities without impairing the temper testing qualities. Thus H with 0·21 per cent. carbon has a tenacity of 33·6 tons and tempers well, as against B with 0·20 per cent. carbon and only 27·4 tons tenacity. Q and D, with 0·127 per cent. carbon have respectively 28·5 and 23·6 tons tenacity; and L with 0·09 per cent. carbon has 29·1 tons tenacity, against 24·1 for A although A has 0·1 per cent. carbon. These examples show that the temper test, unless it is associated with the condition that the

steel must be produced in an acid open hearth, may lead to the use of very dangerous materials, and it is equally possible that it is doing harm by preventing the use of higher tenacity steels. Whether high tenacity steels, free from phosphorus and nitrogen, are suitable for boilers and similar structures, is, of course, a question which can be solved only after the most careful enquiry. . . .

"It would appear, therefore, that at present the only effective means for throwing out material which is not suitable for boilers is to insist that neither Acid nor Basic Bessemer Steel shall be used, and that, whether the steel is made in an Acid or Basic Open-Hearth furnace, the percentage of phosphorus shall not exceed a certain limit, say, 0·06 per cent. If the method of manufacture of a steel plate is not known, it should be tested for nitrogen, and if this exceeds, say, 0·006 per cent., the plate should not be used for a boiler."

9. Cast Steel.—Seating blocks, etc., that have to be fitted to the boiler shell are conveniently made as steel castings. But there is some difficulty in securing sound castings free from blowholes. Important parts such as the headers of water-tube boilers are therefore made of wrought or pressed steel.

10. Nickel Steel.—This material is an alloy of nickel and steel, proportions varying with the tenacity required. Compared with mild steel its tenacity is greater, its ductility is practically the same, and it is much less liable to corrosion. Its hardness and tenacity are little impaired by high temperatures, and hence it is very suitable for the construction of superheater stop valves. From its ductility, durability, and high tenacity it has been largely used in the best boiler practice for smoke and water tubes and plates for high steam pressures.

Experiments carried out by Mr. A. F. Yarrow show that nickel steel containing about 25 per cent. of nickel enables thin-walled tubes to be used for high-pressure work without loss of durability or safety.*

11. Copper.—This material is a reddish ductile metal which

* See Paper by Mr. A. F. Yarrow, *Proc. Inst. Naval Architects*, vol. 41, p. 333, July, 1899.

can be cast, but is seldom used in the form of castings; it is usually rolled into plates and hammered into shape. Copper castings have a tenacity of 10 tons per square inch, sheet copper of about 13·5 tons per square inch. After hammering cold its ductility is reduced and it requires to be annealed. In annealing it should be raised to a temperature of about 500° F., and then quenched in water.

Copper is largely used in boiler work on account of its ductility, resistance to corrosion, and thermal conductivity, principally for smoke tubes and fire-boxes in locomotives. It is also commonly used for the screwed stays in locomotive fire-boxes, and for expansion bends in steel steam piping and steam-pipe ranges on board ship.

The failure of brazed copper steam pipes on board ship, where they are still largely used, has led to a number of serious fatalities, and on several occasions has formed the subject of exhaustive enquiry. As a result of this much greater attention has been paid of late years to the manufacture of copper steam pipes, whether brazed or solid drawn—and the latter are not immune from failure—but failures still occur occasionally resulting in great danger to life and limb, particularly when they occur in closed stokeholds or engine rooms. All brazing operations should be carried out with the utmost care and skill; the spelter used should be quite free from lead; during the brazing a reducing atmosphere, *i.e.* one deficient in oxygen, should be avoided; the brazed pipes should be cooled as rapidly as possible; the braze should be as thin as possible and it should be placed on the top of the pipe to avoid contact with water.

A review of the various failures that have occurred of copper pipes (both brazed and solid drawn) shows that the supposed merits which they possess in virtue of their ductility is accompanied with serious risks, and that when everything is considered mild steel is a much better material.

12. Gun Metal is an alloy of about 90 per cent. copper and 10 per cent. tin. It is largely used for strong castings, being tough and of high tensile strength (about 15 tons per square inch); owing to its expense its use in boiler work is confined to valves and valve seats.

13. Brass is an alloy of copper and zinc in various proportions which range from 70 per cent. of copper and 30 per cent. of zinc to 60 per cent. of copper and 40 per cent. of zinc. An alloy of 2 parts of copper to 1 of zinc is commonly used for locomotive boiler tubes; it is also used for the construction of fittings, valve chests, etc. The tenacity of brass varies from 8 to 13 tons per square inch.

Muntz Metal which can be rolled hot, contains 60 per cent. of copper and 40 per cent. of zinc. It is used for the smoke tubes of locomotives.

Delta Metal is a brass containing a proportion of iron; it can be worked hot or cold, or brazed; it has a high tensile strength, is ductile and withstands corrosion well. Its tenacity, when cast, is about 21 tons per square inch; after forging, about 34 tons per square inch, and this is maintained at high temperatures which ordinary brass will not stand without losing its quality.*

Effect of High Temperatures on the Tenacity of Materials.—In the case of boiler shells and valves exposed to high-pressure steam, the influence of temperature on the strength cannot always be overlooked. Fortunately, in the case of iron and steel at the temperatures ordinarily reached, the strength is not seriously reduced. Most experiments show, for iron and steel, a small gain of tenacity between 60° F. and 500° F., while at temperatures above 600° F. the strength diminishes rapidly with increase of temperature, about 50 per cent. being lost at temperatures of about 1000° F. In the case of copper and the copper alloys a much more marked influence of temperature is observed, these materials beginning to soften at about 600° F., so that they are quite unsuitable for the parts of valves, etc., with which superheated steam comes into contact.

14. Strength of a Boiler Shell or Steam Drum.—For a given steam pressure the thickness of the plates required for a boiler shell or steam drum varies directly as its diameter. When a cylindrical shell is subjected to an internal fluid pres-

* See various Reports of Alloys Research Committee in *Proc. Inst. Mech. Eng.*, 1891–1907.

sure, a tensile stress is set up in the material in directions which are tangential to the perimeter of a transverse section, and this stress is usually called the circumferential or *hoop tension*. The intensity of the hoop tension is a little greater at the inside wall than at the outside, but when the thickness of the plate is small compared to the diameter of the shell the variation is negligible, and the stress may be taken as uniformly distributed.

Let d = internal diameter of the boiler shell in inches,
t = thickness of plate in inches,
p = steam pressure (gauge) in pounds per square inch,
f_1 = hoop tension in the plates in pounds per square inch,
e = efficiency of the longitudinal riveted joints.

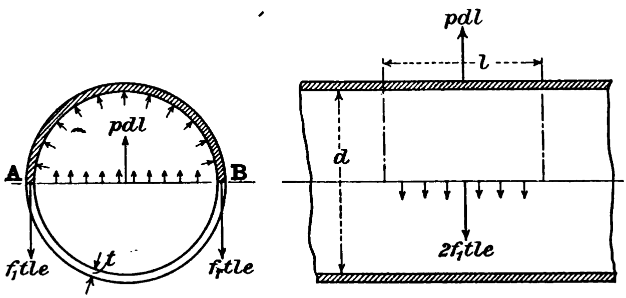


Fig. 13.—Hoop tension in boiler shell.

Consider the equilibrium of the upper half of the cylinder in Fig. 13 of length l. The total hoop tensions perpendicular to the diameter AB will be at A and B.

Intensity of hoop tension × area of metal stressed
$= f_1 \times l \times t \times e$ pounds.

These will balance the resultant fluid pressure across the diametral plane AB, namely

$$p \times d \times l$$

Hence $$p \, . \, d \, . \, l = 2f_1 \, . \, l \, . \, t \, . \, e$$

or $$pd = 2f_1te \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

or $$t = \frac{pd}{2f_1e} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

Longitudinal Tension.—If, as is usual in a steam drum, the ends are connected only by means of the material of the shell, the shell will have, in addition to the hoop tension f_1, a longitudinal tension of intensity, say f_2. The balanced axial forces on any length of the cylinder bounded by a closed end and any normal cross-section CD (Fig. 14) are the axial bursting force $p \times \frac{\pi d^2}{4}$ pounds, and the total longitudinal tension $f_2 \times \pi dt \times e$ pounds. Hence

$$p \times \frac{\pi d^2}{4} = f_2 \times \pi dte$$

or $$pd = 4f_2te \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

or $$t = \frac{pd}{4f_2e} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

From the above it will be seen that the tendency for the shell to burst along a longitudinal seam is twice as great as that along a circumferential seam, or, in other words, for a given diameter of shell d, and internal steam pressure p, the thickness of plate required to withstand safely a given hoop tension is twice as great as that to withstand the longitudinal tension. For this reason it is usual to design the shell from equation (2) and to make the longitudinal joints stronger than the circumferential ones, *i.e.* if the circumferential joints are single riveted, the longitudinal joints will be double riveted.

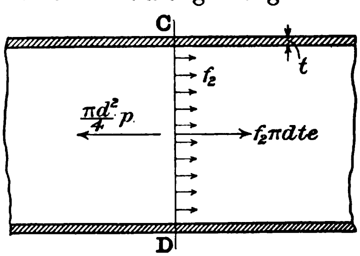


FIG. 14.—Longitudinal tension in boiler shell.

If the cylinder be subjected to an external fluid pressure as is the case with the furnace tubes of the Lancashire boiler, the same equations will hold, but now the material will be subjected to a compressive instead of a tensile stress, and greater care is necessary in construction (see Art. 2, Chap. VII.).

Example.—Estimate the thickness of mild steel plates required (a) for a shell 4 feet diameter and (b) 15 feet diameter.

In each case take the working pressure to be 220 pounds per square inch gauge, the working stress in the plates 10,000 pounds per square inch, and the efficiency of the riveted joints 0·75.

(*a*) From equation (2), $t = \dfrac{220 \times 48}{2 \times 10{,}000 \times 0 \cdot 75}$

$= 0 \cdot 70$, say $\frac{11}{16}$ inch.

(*b*) From equation (2), $t = \dfrac{220 \times 15 \times 12}{2 \times 10{,}000 \times 0 \cdot 75}$

$= 2 \cdot 64$, say $2\frac{5}{8}$ inch.

CHAPTER II

FUELS AND COMBUSTION

1. Solid Fuels.—Coal is formed by the slow decomposition of wood and vegetable matter buried in the earth for a long time and exposed to moisture, great pressure, and the internal heat of the earth. The chemical changes which occur during the formation of coal result in carbon dioxide and marsh gas being set free, these gases being always found in coal-seams. The woody fibre is first changed into *lignite* and *brown* coal, the variety found at Bovey Tracey in Devonshire showing distinct woody structure; other kinds are earthy brown in colour and are friable. Lignite consists of from 50 to 70 per cent. of carbon and 30 to 20 per cent. of oxygen with a large proportion of water.

Bituminous Coal.—The further application of great pressure and decomposition together with the internal heat of the earth changes lignite into *bituminous* or *flaming coals* which may be divided into caking and non-caking varieties.

Caking coal is the most common and burns with a smoky flame. When heated, it swells and softens into a porous pasty mass, and on distillation yields coke.

The non-caking coal of Staffordshire and Lancashire is very brittle and does not cake when heated. It ignites readily and burns freely without softening. *Scotch splint* or *hard coal* is difficult to kindle, but it gives a clear fire at a high temperature. *Cannel coal* is hard and compact, is easy to ignite, and burns with a bright long flame. It is very rich in hydrocarbons and is chiefly used for the manufacture of coal gas. A great many different kinds of bituminous coals are used for steam raising; they contain from 70 to 90 per cent. by weight of carbon.

Welsh *steam coal* gives out a large quantity of heat when burning, forms little or no smoke, and is chiefly used in steam boilers.

Anthracite.—The still further application of great pressure and heat ultimately changes bituminous coal into the hardest coal called *anthracite.* Anthracite which contains about 90 per cent. of carbon, is difficult to ignite, and burns with a short and practically smokeless flame at a high temperature, leaving very little ash. It is chiefly used in suction and other gas plants for the manufacture of producer gas for use in gas engines.

Coke.—In many places, particularly on some railways in Germany, coke is used in preference to coal for steam raising. It is manufactured from raw coal by burning the coal in a limited supply of air in coke ovens with the result that the volatile constituents are driven off and the proportion of carbon increased to about 90 per cent.; it is also an important bye-product from gasworks.

Coke possesses several advantages over coal for steam-raising purposes; it cannot ignite spontaneously, it neither suffers deterioration nor decomposition when exposed to the atmosphere, and it does not produce any smoke. A more uniform temperature is maintained in the furnace; there is very little, if any, tendency to cake, and consequently a uniform rate of evaporation is easily maintained with less variation in the steam pressure. When used for firing locomotives the disadvantages of smoke and sparks are removed, and although it may be slightly more expensive than coal, the above advantages may in many cases make it cheaper than coal from the consumer's point of view.

Other Solid Fuels used for Steam Raising.—In some countries where coal is scarce and too expensive for common use, *wood* is largely used as fuel. When newly felled, timber contains about 50 per cent. of moisture, and even when well-seasoned a considerable amount of moisture remains. It contains more oxygen than coal, and consequently does not require so much air to be supplied for its combustion.

Peat comes halfway between wood and coal; it retains

moisture persistently, but when dry it ignites readily and burns freely, leaving a light grey ash. Well-dried peat contains about 25 per cent. of water, the remaining constituents averaging about 41 per cent. of carbon, 26 of oxygen, 4 of hydrogen and 4 of ash.

"*Magasse*" or "*Bagasse,*" the crushed refuse from sugar-cane factories, is used under boilers on sugar plantations. Like wood, its disadvantage lies in the large amount—over 50 per cent.—of moisture which it contains. It therefore requires to be burnt in a special furnace, and perhaps with the aid of coal or oil fuel. Recently experiments have been made with the bagasse dried in a chamber through which the flue gases were drawn by a fan.*

Town refuse burnt in destructor furnaces is sometimes used to raise steam.

An artificial fuel for which there is an increasing demand, known as *briquette fuel,* is manufactured by mixing coal dust with pitch and compressing the mixture into bricks. The best qualities are equal in heating value to the best Welsh coal, but owing to the pitch used in its manufacture it burns with a smoky flame.

2. Liquid Fuel.—Petroleum consists of a mixture of solid, liquid, and gaseous hydrocarbons obtained either directly as crude petroleum that flows from wells in the earth's crust chiefly in the United States and Russia, or as paraffin oil produced by the distillation of various bituminous coals, principally Scotch shale.

Crude petroleum from the oil wells is a mixture of a great number of solid, liquid, and gaseous hydrocarbons which have very different boiling points. In the process of *refining* the crude oil is separated by fractional distillation into (1) light oils, such as petrol; (2) illuminating oils such as kerosene and russolene; (3) lubricating oils; (4) paraffin wax and vaseline. The residue is a heavy liquid sludge known as *residium* in America and *astatki* in Russia, and is largely used as fuel for steam raising.

* *Engineering*, May 31, 1912.

In addition to petroleum, the liquid hydrocarbon refuse obtained from coal-fed blast furnaces, coke ovens and gasworks has been largely used as fuel in steam boilers. The best known of these oils are, *blast-furnace oil*, *creosote oil*, and *coal-gas tar*, but they are all inferior in heating value to either crude petroleum or astatki.

3. Combustion.—Combustion is a chemical combination of the inflammable constituents of the fuel with oxygen, the process resulting in the evolution of heat. The chemical composition of the fuel being known, the quantity of air required to burn unit weight of the fuel can be calculated as well as its approximate heating or calorific value.

The most common unit of heat used by engineers in estimating the calorific value in this country is the *British Thermal Unit* (B.Th.U.), which may be defined for practical purposes as *the quantity of heat required to raise the temperature of one pound of water* 1° *Fahrenheit.** Another unit of heat which is largely used is the Pound-Degree-Centigrade Unit (C.H.U.), which is the quantity of heat required to raise the temperature of one pound of water 1° Centigrade, one C.H.U. being equal to 1·8 B.Th.U.

The combustible elements in all fuels, both solid and liquid, are carbon, hydrogen, and sulphur. Of these, the sulphur is of minor importance in contributing to the heating value because only small quantities of it are present, and a fuel which contains an appreciable amount of sulphur should be avoided for steam-raising purposes on account of the injurious sulphurous acid gas which it forms on burning.

4. Combustion of Hydrogen.—The process is represented by the equation

$$2H_2 + O_2 = 2H_2O$$

where H_2 represents a molecule of hydrogen whose *atomic* weight is 1, and O_2 a molecule of oxygen whose *atomic* weight

* The correct definition must, strictly speaking, state the part of the scale at which the 1° rise is to be taken, *e.g.* it is often given as 39·1° to 40·1° F., this being the range of maximum density of water. But upon this point there is no definite agreement, and it has little practical importance.

is 16; that is—two molecules of hydrogen and one molecule of oxygen yield two molecules of water vapour or steam.

Experiment also shows that approximately *two volumes of hydrogen burned with* 1 *volume of oxygen yield two volumes of steam at the same temperature and pressure.*

Writing the molecular weight under each of the symbols in the above equation, we have

$$2H_2 + O_2 = 2H_2O \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$
$$2 \times 2 + 32 = 2 \times 18$$
$$4 + 32 = 36$$

or relatively $1 + 8 = 9$, where 8 is the *combining* or *equivalent weight* of oxygen which unites with 1 part by weight of hydrogen.

From these equations we see that—

4 pounds of hydrogen burning to steam (H_2O) yield 36 pounds of steam,

or 1 pound of hydrogen combining with 8 pounds of oxygen yields 9 pounds of steam.

In the process of combustion, 1 pound of hydrogen gives out about 60,930 B.Th.U., but it forms 9 pounds of steam which absorb about $9 \times 970 = 8730$ B.Th.U., leaving in round figures 52,200 B.Th.U. available for useful purposes. Now in actual practice, the steam formed as a result of the combustion of hydrogen in a fuel passes off *as* steam in the flue gases leaving the boiler, hence the effective calorific value of hydrogen must be taken as **52,200 B.Th.U. per pound.** The 60,930 B.Th.U. per pound is called the *Gross* or *Higher Calorific Value,* and the 52,200 B.Th.U. per pound is called the *Nett,* or *Effective* or *Lower Calorific Value.*

Hence the Lower Calorific Value is the Gross Calorific Value *minus* the latent heat of the steam formed during combustion. (The latent heat of steam at atmospheric pressure is approximately equal to 970 B.Th.U. per pound. See Chapter V.)

5. Combustion of Carbon.—For complete combustion when carbon burns to carbon dioxide (CO_2) the action is expressed by the equation—

$$C + O_2 = CO_2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

that is, one atom of carbon and two atoms of oxygen yield one molecule of carbon dioxide.

Writing the relative weights under each of these symbols, we have, since the atomic weight of carbon is 12,

$$\begin{aligned} C + O_2 &= CO_2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2) \\ 12 + 32 &= 44 \end{aligned}$$

or relatively $3 + 8 = 11$, where 3 is the combining or equivalent weight of carbon just as 8 is that of oxygen,

that is, 3 pounds of carbon combining with 8 pounds of oxygen yield 11 pounds of carbon dioxide.

In the process of combustion 1 pound of carbon when burning to CO_2 gives out in round figures **14,500 B.Th.U.**

When 1 pound of carbon burns incompletely forming only carbon monoxide (CO), it only gives out **4400 B.Th.U.,** the action being expressed by the equation

$$\begin{aligned} 2C + O_2 &= 2CO \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3) \\ 24 + 32 &= 2 \times 28 \\ \text{or relatively} \qquad 6 + 8 &= 14 \end{aligned}$$

It will be seen therefore that incomplete combustion of carbon results in a great loss of heat equal to $14,500 - 4400 = 10,100$ B.Th.U. per pound of carbon so burned. Hence the importance of having *no CO* in the flue gases given off from the furnace of a steam boiler.

Again, if CO_2 at a high temperature comes in contact with carbon at a high temperature, the lower oxide CO is formed as represented by the equation

$$CO_2 + C = 2CO \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

This occurs with a thick fire, and if the loss of heat due to the presence of CO in the flue gases is to be prevented air must be admitted *above* the fire to burn the CO to CO_2.

6. Combustion of Sulphur.—The process is represented by the equation

$$S + O_2 = SO_2$$

that is, one atom of sulphur and two atoms of oxygen yield one molecule of sulphur dioxide (SO_2).

Writing the relative weights under each of these symbols, we have, since the atomic weight of sulphur is 32,

$$S + O_2 = SO_2$$
$$32 + 32 = 64$$

or relatively $\quad 8 + 8 = 16$

that is, 8 pounds of sulphur combining with 8 pounds of oxygen yield 16 pounds of SO_2, or, 1 pound of sulphur combining with 1 pound of oxygen gives 2 pounds of sulphur dioxide, and in the process of combustion gives out **4000 B.Th.U.**

Thus the weights of hydrogen, carbon and sulphur combining with 8 pounds of oxygen are respectively, 1, 3, and 8 pounds.

For purposes of reference the important quantities may be summarised as follows :—

Approximate atomic weight.	Molecular weight.	Molecular weight.
H 1	H_2 2	H_2O 18
C 12		CO 28
N 14	N_2 28	CO_2 44
O 16	O_2 32	SO_2 64
S 32		

7. Minimum Quantity of Air required for the Complete Combustion of 1 pound of Solid or Liquid Fuel.— Let 1 pound of the fuel contain,

C pound of carbon.
H pound of hydrogen.
O pound of oxygen.
S pound of sulphur.

From the preceding article, the reactions during combustion may be summarised as follows :—

$$C + O_2 = CO_2 \quad \ldots\ldots\ldots \quad (1)$$
$$12 + 32 = 44$$

or in combining weights, $3 + 8 = 11$

i.e. 1 part by weight of carbon requires $\frac{32}{12} = \frac{8}{3} = 2{\cdot}66$ parts by weight of oxygen; hence 1 pound of carbon requires 2·66 pounds of oxygen for complete combustion.

Next, $$2H_2 + O_2 = 2H_2O \quad . \; . \; . \; . \; . \; . \quad (2)$$
$$4 + 32 = 36$$
or in combining weights, $1 + \ 8 = \ 9$

hence, 1 pound of hydrogen requires 8 pounds of oxygen for combustion.

Lastly, $$S + O_2 = SO_2 \quad . \; . \; . \; . \; . \; . \; . \quad (3)$$
$$32 + 32 = 64$$
or in combining weights, $8 + \ 8 = 16$

hence 1 pound of sulphur requires 1 pound of oxygen for combustion.

Now air contains 23 per cent. by *weight* of oxygen,

hence 100 pounds of air contain 23 pounds of oxygen,
or 1 pound of oxygen is contained in $\frac{100}{23} = 4{\cdot}35$ pounds of air.

Hence :—

1 pound of carbon requires $2{\cdot}66 \times 4{\cdot}35 = 11{\cdot}6$ pounds of air.
1 pound of hydrogen requires $8 \times 4{\cdot}35 = 34{\cdot}8$ pounds of air.
1 pound of sulphur requires $1 \times 4{\cdot}35 = 4{\cdot}35$ pounds of air.

From this we see that 1 pound of the fuel will require

$$\mathbf{11{\cdot}6 \times C + 34{\cdot}8 \times H + 4{\cdot}35 \times S} \textbf{ pounds of air}. \quad (4)$$

Example 1.—The analysis by weight of a certain coal is C 80 per cent., H 5 per cent., S 0·5 per cent.; estimate the theoretical minimum quantity of air required for the complete combustion of 1 pound of the coal.

From (4) we have

Minimum quantity required
$$= 11{\cdot}6 \times 0{\cdot}8 + 34{\cdot}8 \times \tfrac{5}{100} + 4{\cdot}35 \times \tfrac{1}{200}$$
$$= 9{\cdot}28 + 1{\cdot}74 + 0{\cdot}02$$
$$= 11{\cdot}04 \text{ pounds of air.}$$

If only this minimum theoretical quantity of air were

supplied in practice, a certain amount of carbon monoxide would be formed resulting in a large loss of heat as shown in Art. 5. It is, therefore, usual to supply from 1·5 to 2 times the minimum amount necessary for complete combustion in order to obtain the best results from the burning of the fuel. Of course a loss results in heating up this excess air and discharging it up the chimney at a high temperature. For rough calculations when the chemical analysis of the fuel is not known 12 pounds of air per pound of coal may be assumed as the minimum quantity, and since 1 pound of air at atmospheric pressure and 60° Fahrenheit occupies about 13 cubic feet it follows that $12 \times 13 = 156$ cubic feet is the minimum volume of air necessary. Actually from 200 to 300 cubic feet would be allowed in practice, or say, from 16 to 24 pounds of air per pound of coal corresponding respectively to forced and natural draught. To burn one pound of oil fuel completely, the minimum theoretical quantity of air is about 15 pounds of air, but as in the case of coal this must in practice be largely exceeded, and it usually amounts to about the same as for coal.

Example 2.—The analysis of a certain coal by weight is C 85 per cent., H 5 per cent., ash, etc. 10 per cent. If 20 pounds of air are supplied per pound of coal and the combustion is complete, what is the composition of the flue gases by weight?

Total weight of gases per pound of coal

$$= 20 + \text{combustible in 1 pound of coal}$$
$$= 20 + 0{\cdot}9 = 20{\cdot}9 \text{ pounds.}$$

From the chemical formula the proportion of any constituent in a compound is at once found. Thus the proportion of carbon in a given weight of CO_2 is $\frac{12}{44}$. Conversely the weight of CO_2 per 1 pound of carbon is $\frac{44}{12}$, and of H_2O per 1 pound of hydrogen is $\frac{18}{2}$. Hence

Weight of CO_2 per pound of *coal*	$= 0{\cdot}85 \times \frac{44}{12}$	$=$	3·12 pounds.
,, ,, H_2O ,, ,, ,,	$= 0{\cdot}05 \times 9$	$=$	0·45 ,,
,, ,, N_2 ,, ,, ,,	$= 20 \times \frac{77}{100}$	$=$	15·40 ,,
	Total	$=$	18·97 pounds.

∴ weight of O_2 per pound of coal (by difference) is

$20{\cdot}9 - 18{\cdot}97 = 1{\cdot}93$ pounds. The weight of oxygen may also be found by using the method given in Art. 7 as follows :—

$$\text{Weight of oxygen} = 20 \times \tfrac{23}{100} - 0{\cdot}85 \times \tfrac{32}{12} - 8 \times 0{\cdot}5$$
$$= 4{\cdot}6 - 2{\cdot}27 - 0{\cdot}40 = 1{\cdot}93 \text{ pounds.}$$

Hence the flue gases will consist of :—

$$\text{Carbon dioxide } (CO_2) = \frac{3{\cdot}12}{20{\cdot}9} = 0{\cdot}149 \text{ or } 14{\cdot}9 \text{ per cent.}$$

$$\text{Steam } (H_2O) = \frac{0{\cdot}45}{20{\cdot}9} = 0{\cdot}215 \text{ or } 2{\cdot}15 \text{ ,, ,,}$$

$$\text{Nitrogen } (N_2) = \frac{15{\cdot}40}{20{\cdot}9} = 0{\cdot}736 \text{ or } 73{\cdot}6 \text{ ,, ,,}$$

$$\text{Oxygen } (O_2) = \frac{1{\cdot}93}{20{\cdot}9} = 0{\cdot}0925 \text{ or } 9{\cdot}25 \text{ ,, ,,}$$

8. Calculation of the Quantity of Air supplied per pound of Fuel from the Analysis of the Flue Gases.

Let C = percentage of carbon in the fuel *by weight.*
CO_2 = percentage of carbon dioxide in the flue gases *by weight.*
CO = percentage of carbon monoxide in the flue gases *by weight.*
N = percentage of nitrogen in the flue gases *by weight.*
O = ,, ,, oxygen ,, ,, ,, ,, ,, ,,

Considering 100 parts by weight of flue gases there are N parts of nitrogen in it. Now air contains 77 per cent. by weight of nitrogen.

∴ N parts of nitrogen are supplied with $\frac{100}{77} \times N$ parts of air. From (2) and (3) Art. 5 it will be seen that in the 100 parts of flue gases there are

$$(CO_2 \times \tfrac{12}{44}) + (CO \times \tfrac{12}{28}) \text{ parts of carbon.}$$

Hence the ratio of air to carbon is

$$\frac{N \times \frac{100}{77}}{(CO_2 \times \frac{12}{44}) + (CO \times \frac{12}{28})} \quad . \quad . \quad . \quad . \quad (1)$$

Now 1 pound of the fuel contains $\frac{C}{100}$ pounds of carbon.

Hence the weight of air supplied per pound of fuel burned is

$$\frac{N \times \frac{100}{77}}{(CO_2 \times \frac{12}{44}) + (CO \times \frac{12}{28})} \times \frac{C}{100} \text{ pounds} \quad . \quad . \quad (2)$$

which reduces to

$$\frac{\mathbf{N}}{\mathbf{21CO_2 + 33CO}} \times \mathbf{C} \text{ **pounds.**}$$

If CO_2, CO, and N represent the percentages of carbon dioxide, carbon monoxide, and nitrogen respectively in the flue gases *by volume*, then the weight of air supplied per pound of fuel burned is given by

$$\frac{N \times \frac{100}{77} \times 28}{(CO_2 \times \frac{12}{44} \times 44) + (CO \times \frac{12}{28} \times 28)} \times \frac{C}{100} \text{ pounds}$$

where 28 = molecular weight of N_2 and also of CO,
and 44 = molecular weight of CO_2.

The above expression will be found to reduce to

$$\frac{\mathbf{N}}{\mathbf{33(CO_2 + CO)}} \times \mathbf{C} \text{ pounds} \quad . \quad . \quad . \quad (3)$$

9. Loss of Heat by the Flue Gases escaping up the Chimney.

Let s = mean specific heat of the flue gases (about 0·238), *i.e.* the mean heat required to raise unit weight of the mixed gases one degree in temperature,

w = weight in pounds of the flue gases per pound of fuel burned,

t_1 = temperature of the flue gases leaving the boilers (Fahrenheit),

t_2 = temperature of the air in the boiler house (Fahrenheit),

then assuming that there is no preliminary heating of the air supply the heat carried away in the flue gases per pound of fuel is

$$w \times s(t_1 - t_2) \text{ British Thermal Units.}$$

It may here be mentioned that the air supply should as far

as possible be free from moisture, and further that it conduces to economy if it be heated, provided that such preheating is obtained at little cost. Any water vapour present in the air when it passes through the incandescent fuel will become dissociated into its constituent hydrogen and oxygen, and then will again re-combine when they leave the furnace; but they will leave at the higher temperature of the chimney, and the net result is that some heat has been lost in raising the temperature of the water vapour, or expressed in other words, the mean specific heat of the flue gases is higher than if the air had been dry. The loss resulting therefrom is further discussed in Chap. XII., Art. 15. The advantage of a preliminary heating of the air will appear from Art. 6 of Chap. IV.

10. Calculation of the Mean Specific Heat of the Flue Gases.—The calculation requires the analysis of the fuel and also the analysis of the flue gases. The method will be best illustrated by means of a numerical example. Taking the analysis of the fuel by weight as C 88 per cent., H_2 3·6 per cent., O_2 4·8 per cent., other matters (ash, etc.) 3·6 per cent., and the analysis of the dry gases by *volume* as CO_2 10·9 per cent., CO 1·0 per cent., O_2 7·1 per cent., N_2 81 per cent., we proceed as follows :—

The first step in the calculation is to convert the analysis of the dry flue gases *by volume* (the gases are actually always analysed by volume) into the analysis *by weight*.

By multiplying each of the volume proportions by the corresponding molecular weight, adding the products so obtained and then dividing each separate product by the sum of all the products, the proportion by weight of each gas present may be obtained. In the example taken we have :—

Constituent gas.	Analysis by volume.	Molecular weight of gas.
CO_2	10·9% = 0·109	44
CO	1·0% = 0·010	28
O_2	7·1% = 0·070	32
N_2	81·0% = 0·810	28

Following the above method we find :—

$CO_2 = 0{\cdot}109 \times 44 =$	$4{\cdot}796$	$\frac{4{\cdot}796}{30{\cdot}028} = 0{\cdot}159$	proportion by weight
$CO = 0{\cdot}010 \times 28 =$	$0{\cdot}280$	$\frac{0{\cdot}280}{30{\cdot}028} = 0{\cdot}009$	,, ,,
$O_2 = 0{\cdot}070 \times 32 =$	$2{\cdot}272$	$\frac{2{\cdot}272}{30{\cdot}028} = 0{\cdot}076$	,, ,,
$N_2 = 0{\cdot}810 \times 28 =$	$22{\cdot}680$	$\frac{22{\cdot}680}{30{\cdot}028} = 0{\cdot}756$	,, ,,
Total . .	$30{\cdot}028$	$1{\cdot}000$	

Hence the analysis of the dry flue gases by *weight* is CO_2 15·9 per cent., CO 0·9 per cent., O_2 7·6 per cent., N_2 75·6 per cent.

Therefore in 100 pounds of dry flue gases there are

$$15{\cdot}9 \times \tfrac{12}{44} + 0{\cdot}9 \times \tfrac{12}{28} = 4{\cdot}721 \text{ pounds of carbon.}$$

Hence the weight of dry flue gases per pound of dried fuel will be

$$\frac{100 \times 0{\cdot}88}{4{\cdot}721} = 18{\cdot}64 \text{ pounds.}$$

But the 0·036 pound of hydrogen in the fuel produces $0{\cdot}036 \times 9 = 0{\cdot}324$ pound of H_2O in the form of steam; hence the actual weight of each constituent in the flue gases due to 1 pound of dry fuel may be determined as follows :—

Constituent.	Weight in pounds per pound of dry fuel burned.	Weight in pounds of each constituent in 1 pound of flue gases.
CO_2	$18{\cdot}64 \times 0{\cdot}159 = 2{\cdot}964$	$\frac{2{\cdot}964}{18{\cdot}964} = 0{\cdot}156$
CO	$18{\cdot}64 \times 0{\cdot}009 = 0{\cdot}168$	$\frac{0{\cdot}168}{18{\cdot}964} = 0{\cdot}009$
O_2	$18{\cdot}64 \times 0{\cdot}076 = 1{\cdot}417$	$\frac{1{\cdot}417}{18{\cdot}964} = 0{\cdot}075$
N_2	$18{\cdot}64 \times 0{\cdot}756 = 14{\cdot}091$	$\frac{14{\cdot}091}{18{\cdot}964} = 0{\cdot}742$
H_2O	$= 0{\cdot}324$	$\frac{0{\cdot}324}{18{\cdot}964} = 0{\cdot}018$
	Total . . $18{\cdot}964$	$1{\cdot}000$

The specific heats at constant pressure of the above gases may be taken as :—

$$
\begin{aligned}
CO_2 &= 0{\cdot}216\\
CO &= 0{\cdot}245\\
O_2 &= 0{\cdot}218\\
N_2 &= 0{\cdot}244\\
H_2O &= 0{\cdot}480
\end{aligned}
$$

Hence the mean specific heat of the flue gases is :

(weight of $CO_2 \times 0{\cdot}216$)+(weight of $CO \times 0{\cdot}245$)+(weight of $O_2 \times 0{\cdot}218$)+(weight of $N_2 \times 0{\cdot}244$)+(weight of $H_2O \times 0{\cdot}480$).

In the above example this becomes,

$$
\begin{aligned}
&(0{\cdot}156 \times 0{\cdot}216)+(0{\cdot}009 \times 0{\cdot}245) + (0{\cdot}075 \times 0{\cdot}218)\\
&+(0{\cdot}742 \times 0{\cdot}244) + (0{\cdot}018 \times 0{\cdot}480)\\
&= 0{\cdot}2417 = 0{\cdot}242 \text{ practically.}
\end{aligned}
$$

11. Approximate Method of Calculating the Weight of Air supplied per pound of Fuel from the Analysis of the Flue Gases.—The excess of air supplied over the minimum theoretical quantity required for complete combustion may be obtained from the well-known formula

$$
\left.\begin{array}{l}\text{Ratio of total air supplied to minimum}\\ \quad\text{theoretical quantity . . . . . .}\end{array}\right\} = \frac{21}{21 - \dfrac{79 \times O_2}{N_2}}
$$

where O_2 = percentage *by volume* of oxygen in the flue gases.
N_2 = percentage *by volume* of nitrogen in the flue gases.

This formula may be deduced as follows :—

By volume air consists of 21 parts of oxygen and 79 parts of nitrogen. If, therefore, excess oxygen O_2 is found in the flue gases, the excess nitrogen that must have accompanied it is $\frac{79}{21} O_2$. But the ratio

$$
\frac{\text{total air supplied}}{\text{min. theoretical quantity}} = \frac{\text{nitrogen in flue gases}}{\text{nitrogen in flue gases} - \text{excess nitrogen}}
$$

$$
= \frac{N_2}{N_2 - \dfrac{79}{21} O_2}
$$

$$= \frac{1}{1 - \frac{79}{21} \times \frac{O_2}{N_2}}$$

$$= \frac{21}{21 - \frac{79 \times O_2}{N_2}}$$

The amount of air actually supplied will be this ratio multiplied by the minimum theoretical quantity.

Since no very great error is made by assuming 12 pounds of air per pound of *coal* as the minimum theoretical amount required for complete combustion, this method is very convenient for approximate values when the chemical analysis of the *coal* is unobtainable.

12. Calorific Value of Fuels.—In calculating the calorific value of a solid or liquid fuel Dulong's formula may be used. In this formula it is assumed that all the oxygen present in the fuel is already combined with one-eighth of its weight of hydrogen in the proportion to form water so that the hydrogen *available* for combustion is only $H - \frac{O}{8}$. Using the same notation as in Art. 7, the lower calorific value of the fuel as calculated by this formula * is

$$\mathbf{14{,}500\,C + 52{,}200\left(H - \frac{O}{8}\right) + 4000\,S \quad B.Th.U. per lb.} \quad (1)$$

The effective calorific value of a fuel can only be determined *accurately* by direct experiment in an approved calorimeter, the most reliable instrument being the "Bomb" calorimeter.

It will be found that the above formula invariably gives a result too low as compared with the value obtained by direct calorimetric tests, probably because more hydrogen is free than is assumed. Also that the same formula will not be equally correct for both solid and liquid fuels on account of the complex composition of the hydrocarbons in oil fuels. The Author found † that for the steam coals used in practice the formula—

$$\mathbf{14{,}400\,C + 52{,}200\,H \qquad B.Th.U. per pound}$$

* This formula gives the lower calorific value, the higher calorific value would be given by $14{,}500\,C + 60{,}930\left(H - \frac{O}{8}\right) + 4000\,S$.

† See *Engineer*, Feb. 17, 1911.

gives results for the lower calorific value very closely in agreement with those obtained in the "Bomb" calorimeter, whilst for oil fuels the formula

$$\mathbf{13{,}500\ C + 52{,}200\ H \qquad B.Th.U.\ per\ pound}$$

gave results practically the same as those given by the Bomb Calorimeter. Numerous other formulæ have been given from time to time all more or less accurate, but it should be remembered that when accurate results are required recourse must be made to the Bomb Calorimeter. The accompanying Tables III. and IV. (pp. 40 and 41) give the average chemical analysis and calorific value of various solid and liquid fuels used in practice.

The "evaporative power" is simply the calorific value divided by 966, the divisor (966) being taken as the latent heat of 1 pound of steam at 212° F. It is, therefore, the number of pounds of water that can be evaporated from and at 212° F. by 1 pound of the coal in question.

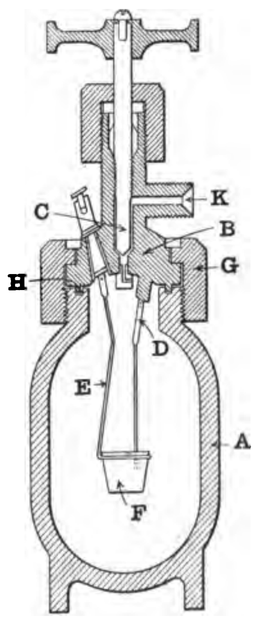


FIG. 15.—The Bomb Calorimeter.

13. The Bomb Calorimeter.—The latest form of the Mahler-Cook Bomb is shown in section in Fig, 15. The steel bomb A, the inside face of which is coated with a special enamel to protect the steel against corrosion, is made with spherical ends in order to withstand the high pressure to which it is subjected. The cover B carries a needle valve C for admitting oxygen under pressure into the bomb. A stout platinum wire support D is screwed directly into the under face of the cover, the other support E to the crucible F being attached to an insulated plug passing through the cover as shown. The platinum crucible F, containing the fuel to be tested, is carried in a stirrup at the

TABLE III.

Composition and Calorific Value of Various Solid Fuels.

Description of fuel.	Moisture, per cent.	Carbon, per cent.	Hydrogen, per cent.	Oxygen + nitrogen, per cent.	Sulphur, per cent.	Higher calorific value.	
						Calories, per gramme.	B.Th.U. per pound.
Wood (ordinary) .	28·9	36·4	4·6	29·6	—	3310	5,960
Wood (dried) . .	6·9	47·4	5·6	41·0	—	4480	8,060
Peat (poor) . . .	20·8	40·8	3·3	31·4	—	3770	6,790
Peat (air dried) .	6·1	53·2	5·5	35·0	—	5490	9,880
Cannel coal (Wigan)	0·6	78·4	5·1	11·20	0·4	7760	13,970
Cannel coal (Scotch)	4·0	75·42	6·18	9·98	2·18	7500	13,550
Yorkshire caking coal	2·20	84·10	4·93	7·00	0·55	7420	13,370
Durham caking coal	1·14	84·34	5·30	6·00	0·78	8300	14,870
Newcastle steam coal	1·20	81·30	5·30	9·90	1·20	8160	14,690
Durham steam coal	0·80	81·50	4·60	6·00	1·20	7970	14,350
Welsh steam coal .	—	83·80	4·80	5·10	1·40	8050	14,490
Nixon's navigation steam coal . .	1·00	87·80	4·10	5·00	—	8580	15,450
Slack coal . . .	7·30	67·90	4·90	16·00	1·30	7220	13,000
Welsh anthracite .	—	91·50	3·50	3·40	0·60	8460	15,220
American ,, .	3·40	86·40	2·00	2·20	—	7480	13,470
Average English coke	4·80	88·40	1·40	3·27	0·35	7600	13,600

TABLE IV.

*Composition and Calorific Value of Various Liquid Fuels.**

Description of fuel.	Specific gravity.	Carbon, per cent.	Hydrogen, per cent.	Oxygen + nitrogen, per cent.	Sulphur, per cent.	Higher calorific value. Calories, per gramme.	Higher calorific value. B.Th.U. per pound.
Russolene (H.Y.O.)	0·890	85·95	13·50	—	0	10,900	19,600
American kerosene	0·780	85·05	14·40	—	0	11,160	20,100
American royal daylight . . .	0·797	85·70	14·20	—	0	11,170	20,100
American petrol .	—	80·58	15·10	4·31	0	11,080	19,950
Refined Russian petroleum (Baku) .	0·825	86·00	14·00	—	—	11,270	20,290
American crude petroleum . .	—	86·90	13·10	—	—	10,910	19,650
Russian crude caucasian (Novorossisk)..	—	84·90	11·63	1·46	—	10,330	18,600
Baku heavy oil . .	—	86·70	12·94	—	—	10,800	19,450
Naphtha	—	75·57	10·57	3·91	—	9,250	16,650
Russian crude . .	0·871	86·90	13·10	—	0	10,830	19,500
Java crude . . .	0·867	87·10	12·70	—	0	10,650	19,190
Canadian crude .	0·859	86·92	12·87	—	0·35	10,800	19,450
Texas crude . . .	0·947	86·62	11·80	—	0·63	10,520	18,960
Solar oil	0·896	86·61	12·60	—	0·30	10,780	19,430
Coal oil	0·917	83·20	11·87	—	1·56	10,220	18,410

* From a paper by the author on "The Calorific Values of Solid and Liquid Fuels."—*The Engineer*, Feb. 17, 1911.

end of the wire E, and a fine platinum ignition wire with its ends firmly tied round the supports D and E dips down into the fuel which is thereby ignited when the wire is made red hot by the passage of an electric current. The cover is held in position by a large hexagonal nut G, a gas tight joint being made by the spigot H and a lead ring in a recess in the top of the bomb.

When in use the bomb is placed inside a copper calorimeter vessel containing a known weight of water, which is itself enclosed in an outer double-walled vessel. The outer vessel is well lagged on the outside with a thick layer of felt and is surmounted by a framework carrying a stirring gear for the water in the calorimeter.

The apparatus is suitable for the direct determination of the higher calorific value of both solid and liquid fuels.

Method of Using.—About 1 gram of the fuel is introduced into the crucible which is then placed in the stirrup and the ignition wire attached as explained above. The cover is then placed in position on the bomb and fixed there by screwing down the hexagonal nut G. Oxygen at a pressure of 25 atmospheres is then admitted through a nut and cone attachment at K by opening the valve C. When full of oxygen at this pressure the valve is closed, wires from an electric circuit fixed in the outside terminals attached to the supports E and D, and the bomb placed in water contained in the calorimeter. The amount of water should be such that it covers about half of the hexagonal nut G; in any case it should be such that the ignition wire is not short-circuited, otherwise it would be impossible to fire the fuel.

A delicate "Bechmann" thermometer graduated in $\frac{1}{100}$° C. with a range of about 5° is placed in the water which is then stirred by means of the stirring gear. On the temperature becoming approximately steady, readings of the temperature are taken every minute for, say, five minutes, after which the fuel is fired.*

* A battery of e.m.f. about 12 volts will be found suitable for this purpose; or if any electric light circuit is available the firing can be satisfactorily carried out by the current passing through a 16 or 32 c.p. lamp, the requisite connections being simple to arrange.

The temperature will rapidly rise and in from 5 to 10 minutes will attain a maximum value, after which it will commence to fall slowly. When the fall of temperature has become regular, the readings are stopped and the experiment is completed.

As soon as possible after each experiment the bomb is removed from the calorimeter, the gases let off and the bomb opened; traces of nitric acid will be found, which, if allowed to remain inside, will attack the valve spindle C and any portion of the metal surface which becomes exposed. To prevent this occurring the bomb should be washed out after the experiment with dilute caustic soda, after which it should be dried with a cloth and an open bottle containing caustic soda or lime placed inside the bomb when not in use in order to keep it dry by absorbing any moisture present.

The method of calculating the calorific value will be best illustrated by the following example of an actual experiment. The true rise of temperature due to the combustion of the fuel may, however, be determined by means of a graph connecting temperature and time, extrapolating to determine the maximum temperature which would have been attained had no subsequent cooling occurred.

Weight of coal taken	= 1·035 grams.
weight of water used in calorimeter	= 2050 c.cs. = 2050 grams.
water equivalent of calorimeter	= 750 grams.
∴ total equivalent of water	= 2050 + 750 = 2800 grams.

Time. hr.	min.	*Temperature.*	*Time.* hr.	min.	*Temperature.*
2	30	0·64	2	41	2·92
2	31	0·643	2	42	2·93
2	32	0·645	2	43	2·935
2	33	0·645	2	44	2·935 (max.)
2	34	0·645 (fired)	2	45	2·933
2	35	1·360	2	46	2·930
2	36	2·340	2	47	2·925
2	37	2·50	2	48	2·920
2	38	2·70	2	49	2·915
2	39	2·85	2	50	2·910
2	40	2·90			

From the above readings we see that the observed rise of temperature is $2{\cdot}935 - 0{\cdot}645 = 2{\cdot}29^\circ$ C.

The rate of fall in temperature from 2–44 to 2–50 is $\frac{0{\cdot}02^\circ}{6}$ per minute approximately.

The temperature is rising from 2–34 to 2–44, *i.e.* for a period of 10 minutes, hence the temperature correction to be added to the observed rise is

$$\frac{0{\cdot}02}{6} \times 10 = 0{\cdot}033^\circ \text{ C.}$$

Hence the true rise of temperature $= 2{\cdot}29 + 0{\cdot}033 = 2{\cdot}323^\circ$ C.

Heat given out by the combustion of 1·035 grams of coal

$$= 2800 \times 2{\cdot}323 \text{ calories}$$

$$\text{Calorific value of the coal} = \frac{2800 \times 2{\cdot}323}{1{\cdot}035}$$

$$= 6280 \text{ calories per gram.}$$

or 6280 C.H.U. per pound.

To convert from calories per gram to B.Th.U. per pound we only have to multiply by 1·8, hence the calorific value may also be stated as

$$6280 \times 1{\cdot}8 = 11{,}300 \text{ B.Th.U. per pound.}$$

Instead of fine platinum wire being used for the ignition of the fuel, fine iron wire might be used. This is not to be recommended, however, on account of the danger of damage being done to the enamel lining of the bomb; in addition, a correction would have to be made for the heat given out by the combustion of the iron wire. Nickel wire, which is non-combustible, might also be used instead of platinum if desired.

The usual method of determining the water equivalent of the calorimeter is to burn in the bomb a certain weight of a substance whose calorific value is known, *e.g.* naphthaline of calorific value 9688 calories per gram is suitable for this purpose. The experiment is carried out in exactly the same way as described above, and from the true rise in temperature the water equivalent is calculated.

14. Preparation of a Sample of Coal.—The sample may be taken in the mine from the coal seam itself, from a truck or

wagon, or from a heap, and the amount taken should never be less than about 20 pounds for use in the laboratory. In the mine the usual method is to take a complete vertical section of the seam, rejecting only such parts of shale or stony matter as would be rejected commercially. If the sample be taken from a wagon or heap, the operation becomes somewhat more difficult. In taking a sample from a wagon the door should be opened and portions taken 1 foot apart from the top, centre and bottom, but it is advisable before doing so, to note the amount of each band present, and then select the proportions proportionately. A method similar to this is adopted should the coal be in a heap, only the parts selected can be more widely spaced. If in any case the total sample exceeds 20 pounds, which is almost certain to be the case when taken from a wagon or heap, the whole is crushed to pieces about the size of walnuts, mixed thoroughly, spread out in a round flat heap and then quartered. From this, two opposite quarters are taken, and the same process repeated with these until the desired amount is obtained.

When taken to the laboratory, the 20 pound sample is reduced still further by being crushed through a $\frac{1}{4}$-inch sieve. As above it is then thoroughly mixed, spread out and quartered, the opposite quarters being again crushed to pass through an $\frac{1}{8}$-inch sieve. A number of 100 gram samples are taken from this and placed in tightly stoppered bottles to prevent access of air. The final sample to be tested in the "Bomb" is crushed to a fine powder before being used.

Example 1.—The analysis of the coal used in a boiler trial was C 88 per cent., H_2 3·6 per cent., O_2 4·8 per cent., other matters 3·6 per cent. The flue gases had an analysis by volume of CO_2 10·9 per cent., CO 1 per cent., O_2 7·1 per cent., and N_2 81 per cent. Estimate the weight of flue gases per pound of coal burnt, and the probable temperature of combustion.

From (3) Art. 8 we have:

$$\left.\begin{array}{l}\text{Air supplied per pound of}\\ \quad\text{coal}\end{array}\right\} = \frac{N}{33(CO_2 + CO)} \times C$$

$$= \frac{81}{33 \times 11{\cdot}9} \times 88 = 18{\cdot}15 \text{ pounds.}$$

$$\left.\begin{array}{l}\text{Combustible in 1 pound of}\\ \quad\text{coal}\end{array}\right\} = 0{\cdot}88 + 0{\cdot}036 = 0{\cdot}91 \text{ practically.}$$

$$\left.\begin{array}{l}\text{Weight of gases per pound}\\ \quad\text{of coal}\end{array}\right\} = 18{\cdot}15 + 0{\cdot}91 = 19{\cdot}06 \text{ pounds,}$$

which agrees closely with the value 18·96 obtained by a different method in Art. 10.

The calorific value of the coal may be approximately estimated by using Dulong's formula (1) Art. 12, namely

$$\begin{aligned}\text{Calorific value} &= 14{,}500 \times 0{\cdot}88 + 52{,}200\left(0{\cdot}036 - \frac{0{\cdot}048}{8}\right)\\ &= 12{,}760 + 1566\\ &= 14{,}326, \text{ say, } 14{,}330 \text{ B.Th.U. per pound.}\end{aligned}$$

Now 1 pound of coal produces 19·06 pounds of flue gases having a mean specific heat of 0·242 (see Art. 10); hence the probable temperature of combustion above the temperature of the air in the boiler house will be

$$\frac{14{,}330}{19{\cdot}06 \times 0{\cdot}242} = 3100^\circ \text{ F.}$$

It should be noticed that the greater the quantity of air supplied, the lower will be the theoretical temperature of combustion.

Example 2.—Estimate the quantity of heat lost by incomplete combustion of the coal in Example 1.

The heat lost by incomplete combustion will be equal to the weight of carbon burned to CO per pound of fuel burned multiplied by the difference between the calorific values of carbon when burned to CO_2 and when burned to CO.

The proportion of carbon burned to CO is found as follows:

$$\left.\begin{array}{l}\text{Carbon in 10·9 parts by}\\ \quad\text{volume of } CO_2 \quad . \quad .\end{array}\right\} = 10{\cdot}9 \times \tfrac{12}{44} \times \text{molecular weight of } CO_2$$

$$= 10{\cdot}9 \times \tfrac{12}{44} \times 44 = 130{\cdot}8 \text{ parts.}$$

$$\left.\begin{array}{l}\text{Carbon in 1 part by}\\ \quad\text{volume of CO} \; . \quad . \quad .\end{array}\right\} = 1 \times \tfrac{12}{28} \times \text{molecular weight of CO}$$

$$= \tfrac{12}{28} \times 28 = 12 \text{ parts.}$$

Total carbon in gases $= 130{\cdot}8 + 12 = 142{\cdot}8$ parts,

hence proportion of carbon burned to CO $= \dfrac{12}{142{\cdot}8} = 0{\cdot}084$ or

8·4 per cent., and the heat lost per pound of coal through incomplete combustion is

$$0{\cdot}88 \times 0{\cdot}084(14{,}500 - 4400) = 746{\cdot}6 \text{ B.Th.U.}$$

which is $\frac{746{\cdot}6}{14{,}330} = 0{\cdot}052$ or 5·2 per cent. of the heat in the coal.

Example 3.—In a boiler trial the fuel analysis, dry coal as burned, was C 85 per cent.; H_2 4 per cent.; O_2 7 per cent.; ash, etc., 4 per cent.; and the flue gas analysis by weight was CO_2 11 per cent.; CO 1·5 per cent.; O_2 7·1 per cent.; N_2 80·4 per cent. The temperature of the flue gases leaving the boiler was 600° F, and the boiler house temperature was 70° F. Estimate :—

(*a*) The proportion of carbon burned to CO and the heat lost through this imperfect combustion expressing the latter as a percentage of the heat in the fuel.

(*b*) The heat carried away in the flue gases per pound of coal burned, the mean specific heat of the gases being taken as 0·24.

(*a*) Carbon in 11 parts by weight of $CO_2 = 11 \times \frac{12}{44} = 3{\cdot}00$ parts
Carbon in 1·5 parts by weight of $CO = 1{\cdot}5 \times \frac{12}{28} = \underline{0{\cdot}64}$ „
Total $= \underline{3{\cdot}64}$ „

Proportion of carbon burned to CO $= \frac{0{\cdot}64}{3{\cdot}64} = 0{\cdot}175$ or 17·5 per cent.

Heat lost per pound of fuel through imperfect combustion $= 0{\cdot}85 \times 0{\cdot}175\ (14{,}500 - 4400) = 1502$ B.Th.U.

$$\text{Calorific value of the fuel} = 0{\cdot}85 \times 14{,}500 + 52{,}200\left(0{\cdot}04 - \frac{0{\cdot}07}{8}\right)$$
$$= 12{,}325 + 1630$$
$$= 13{,}955 \text{ B.Th.U. per pound.}$$

Hence proportion lost through incomplete combustion $= \frac{1502}{13{,}955} = 0{\cdot}1077$ or 10·77 per cent.

(*b*) Air supplied per pound of coal by (2) Art 8 is

$$\frac{80 \cdot 4}{(21 \times 11) + (33 \times 1 \cdot 5)} \times 0 \cdot 85$$

$$= \frac{80 \cdot 4}{231 + 49 \cdot 5} \times 0 \cdot 85$$

$$= \frac{80 \cdot 4 \times 0 \cdot 85}{280 \cdot 5} = 24 \cdot 38 \text{ pounds.}$$

Combustible in 1 pound of coal = 0·85 + 0·04 = 0·89 pound.

Weight of flue gases per pound of coal = 24·38 + 0·89 = 25·27 pounds.

Hence, heat carried away by the flue gases per pound of coal is (Art. 9)

$$25 \cdot 27 \times 0 \cdot 24(600 - 70) = 3214 \text{ B.Th.U.}$$

$$= \frac{3214}{13{,}955} = 0 \cdot 2305 \text{ or } 23 \cdot 05 \text{ per cent.}$$

Examples on Chapter II.

1. The analysis of a certain coal by weight is C 84·0 per cent.; H 5·6 per cent.; ash 3·0 per cent.; other matters 7·4 per cent. If 25 pounds of air are supplied per pound of coal and the combustion is complete, estimate the analysis of the flue gases by weight.

Ans. CO_2 = 11·90 per cent.; H_2O = 1·90 per cent.; N_2 = 74·30 per cent.; O_2 = 11·90 per cent.

2. The volumetric analysis of the *dry* flue gases from a boiler is: CO_2 = 8·2 per cent.; CO = nil; O_2 = 11·2 per cent.; N_2 = 80·6 per cent. Convert this into an analysis by weight.

Ans. CO_2 = 12·1 per cent.; O_2 = 12·0 per cent.; N_2 = 75·9 per cent.

3. If in Question 2, the analysis of the coal is C = 84 per cent.; H = 5·6 per cent.; ash = 3·0 per cent.; oxygen and other matters (by difference) = 7·4 per cent., estimate the mean specific heat of the flue gases leaving the boiler. Take the specific heats at constant pressure of the above gases as

$$O_2 = 0 \cdot 218$$
$$CO_2 = 0 \cdot 216$$
$$H_2O = 0 \cdot 480$$
$$N_2 = 0 \cdot 244$$

Ans. 0·246.

4. The analysis of the coal used in a boiler trial was C = 79·7 per cent.; H = 4·9 per cent.; ash = 4·1 per cent.; other matters 11·3 per

cent. The flue gases had an analysis by volume of CO_2 = 8·3 per cent.; O_2 = 11·4 per cent.; N_2 (by difference) = 80·3 per cent. Estimate the total weight of flue gases and the weight of excess air per pound of coal burned.

Ans. 24·25 pounds; 12·5 pounds.

5. In a boiler trial the fuel analysis, dry coal as burned, was C = 86 per cent.; H_2 = 4 per cent.; ash = 3 per cent.; other matters (by difference) = 7 per cent. The flue gas analysis by weight was CO_2 = 13·5 per cent.; CO = 1 per cent.; O_2 = 6·1 per cent.; N_2 = 79·4 per cent. The temperature of the flue gases leaving the boiler was 600° F., and the boiler-house temperature was 60° F. Estimate:

(*a*) The proportion of carbon burned to CO and the heat lost through imperfect combustion, expressing the latter as a percentage of the heat in the coal.

(*b*) The total heat carried away in the flue gases per pound of coal burned, the mean specific heat of the gases being taken as 0·25.

Take the calorific value of carbon when burned to CO_2 as 14,500 B.Th.U. per pound, and when burned to CO as 4400 B.Th.U. per pound, and the lower calorific value of hydrogen as 52200 B.Th.U. per pound.

Ans. (*a*) 0·104 or 10·4 per cent.; 903 B.Th.U. or 6·2 per cent.,
(*b*) 3033 B.Th.U.

6. The following data were obtained as the result of a boiler trial: Analysis of coal, C = 85 per cent.; H = 4 per cent.; ash = 5 per cent.; other matters = 6 per cent.

Analysis of flue gases by volume, CO_2 = 12 per cent.; CO = 1·5 per cent.; O_2 = 8·0 per cent.; N_2 = 78·5 per cent.

Temperature of flue gases leaving boiler = 550° F.

Temperature of air in boiler house = 50° F.

Calorific value of ashes = 980 B.Th.U. per pound.

Calorific value of coal = 14,400 B.Th.U. per pound.

Mean specific heat of products of combustion = 0·25, specific heat of air = 0·238.

Estimate as a percentage of the heat in the coal:—

(*a*) Heat carried away by products of combustion.

(*b*) Heat carried away by excess air.

(*c*) Heat lost by incomplete combustion.

(*d*) Heat lost by unburnt carbon in ash.

Ans. (*a*) 1517 B.Th.U. = 10·5 per cent.
(*b*) 443 ,, = 3·0 ,, ,,
(*c*) 954 ,, = 6·6 ,, ,,
(*d*) 49 ,, = 0·3 ,, ,,

CHAPTER III

BOILER FURNACES

1. General Considerations.—In the previous chapter the theory of combustion was considered; we now proceed to consider the methods adopted in practice in order to get as much as possible of the heat in the fuel into steam ready to do useful work in the steam-engine's cylinder. In the conversion of the heat contained in a fuel into a form in which it can do useful work, there are always losses, and a steam boiler is designed in such a manner that these losses are reduced to a minimum. A large proportion of these losses takes place in the furnace and the design of a good furnace is by no means an easy matter.

The losses which take place are due to:—

(1) Incomplete combustion owing to the unavoidable admission of too much, or too little air to the furnace.

(2) Radiation from the furnace and boiler flues.

(3) Infiltration of air through the brickwork setting of the boiler.

(4) The effect on combustion of the gases coming into contact with the comparatively cool heating surface of the boiler, which reduces the temperature of the gases before complete combustion is attained.

(5) The heat carried off by the flue gases escaping up the chimney.

The first of the above losses can be minimised to a great extent by paying careful attention to stoking or by various mechanical means. The second loss must always exist, but is not as a rule very serious in a well-designed boiler, whilst

with well-built brickwork the third loss can be practically prevented.

The fourth of the above losses is one of the most difficult to prevent, and constitutes the greatest difficulty with which the boiler designer has to contend. It arises from the varying temperatures at which the volatile gases are liberated due to differences in the chemical composition and physical properties of various fuels, and the varying effect of the cool heating surface on these gases. This explains why, in practice, the same useful effect cannot be obtained from bituminous as from anthracite coals, notwithstanding the fact that these coals may have very nearly the same heating value.

With anthracite coal, disintegration and distillation take place very slowly, with Welsh steam coal somewhat faster, and with bituminous coal practically simultaneously with its being thrown into a boiler furnace.

Although with anthracite and coke a certain space in the furnace is required for combustion before the gases come in contact with the cool boiler heating surface, the underlying principle necessary to obtain proper combustion of these fuels is to proportion the rate at which the fuel is burned to the rate at which air can be supplied. In other words, the highest efficiency obtainable with these fuels is chiefly dependent upon the rate of combustion, and to ensure a certain quantity being burned in an existing boiler furnace, either the grate area must be increased beyond the size required for the same quantity of bituminous coal, or the intensity of draught must be increased. Thus, when burning anthracite or coke in a furnace proportioned for bituminous coal, either an extra high chimney is required, or an artificial method of increasing the draught commonly called *forced* or *induced* draught must be used.

The fifth loss is reduced to a minimum in two ways, namely, by supplying the minimum quantity of air which will give complete combustion, and secondly, by cooling the gases down to as low a temperature as possible before discharging them up the chimney. If ordinary chimney draught is used there is a limit to the temperature to which the gases may be reduced, because the lower the temperature, the less is the

draught produced by the same height of chimney. With induced or forced draught, however, there is no reason why the gases should not be cooled down to very nearly the same temperature as the steam in the boiler by using an economiser, since in this case the draught being produced by mechanical means is independent of the temperature of the gases.

2. Design of Furnaces.—It will be seen from Art. 1 above that the design of a boiler furnace must be suited to the nature of the fuel and also to the type of boiler and to the means employed for stoking, *i.e.* by hand or mechanically. Boiler furnaces may be classified into two types, *internally fired* and *externally fired* furnaces. Internally fired furnaces include those which form part of the boiler itself, as in Lancashire and Cornish boilers, in some types of marine boilers, and in locomotives. Externally fired furnaces are those which are external to the boiler itself as in water-tube boilers. Lancashire boilers and their modifications are also sometimes fitted with special externally fired furnaces for burning refuse fuels.

The best ratio of heating surface to grate area depends upon the type of boiler to which the furnace is fitted. On account of the differences in the furnace conditions, the rapidity of water circulation, and the speed with which the hot gases move over the heating surface, a ratio suitable for a Lancashire or Cornish boiler would be altogether unsuitable for a water-tube boiler or a locomotive boiler. Table I., page 4, which is compiled from actual boilers shows the ratios commonly adopted with various types of boilers.

When designing a furnace to have a given grate-area, the relation to adopt between the length and width of the grate will again depend upon whether the furnace is to be internally or externally fired, and also whether the firing is to be done by hand or by mechanical stokers. If hand firing is adopted, the length of the grate should not be excessive; otherwise considerable difficulty will be experienced both in firing and in cleaning the grate.*

* Difficulty will be experienced, even by a skilful stoker, in performing these operations if the length of the grate exceeds about 7 feet.

With Lancashire and similar types of boilers, the allowable width of grate will be fixed by the diameter of the furnace tube which is itself dependent upon the diameter of the boiler shell. The greatest width of grate permissible is about 4 feet, corresponding to a Lancashire boiler about 9 feet in diameter, whilst the least width is about 2 feet 6 inches, suitable for a Cornish boiler about 5 feet in diameter.

Externally fired boilers impose no such restrictions on the width of grate, but, when hand fired, efficient stoking and cleaning of the grate can only be ensured by fitting the requisite number of furnace doors to the furnace, many large water-tube boilers having four or more doors.

3. Firebars and Width of Air-Spaces.—A very large number of different designs of firebars have been proposed and tried from time to time, but the most common types in use are made of cast iron and vary from $2\frac{1}{2}$ to 4 feet in length, from $3\frac{1}{2}$ to 5 inches in depth, with air-spaces between them of $\frac{1}{2}$ to $\frac{3}{4}$ inch and a thickness at the top of from $\frac{3}{4}$ to 1 inch tapering to $\frac{5}{8}$ to $\frac{7}{8}$ inch at the bottom. The Admiralty use wrought iron or steel bars about $\frac{3}{8}$ inch thick with air-spaces of from $\frac{1}{4}$ to $\frac{3}{8}$ inch. Such bars as the above are found suitable for both natural chimney draught and also for a moderately forced draught.

The best width for the air-space between adjacent firebars is largely dependent upon the size and class of fuel used and the intensity of draught employed. For any particular fuel and draught practical experience can alone determine the best ratio of width of air-space to width of bar. A total air-space between the bars of about one-third the area of the grate is given roughly by dimensions of $\frac{1}{2}$ inch of air and 1 inch of bar, and this with natural draught yields good results, but with a strong mechanically produced draught the air-spaces may be considerably reduced. In the latter case they may be made so narrow that the resistance offered to the passage of the air through the fire is very much less than that through the air-spaces; for all practical purposes the total resistance is then independent of the thickness of the fire, which is in itself an advantage, particularly when it is remembered that the same rate of combustion may be maintained as with natural draught and wide air-spaces.

Again with narrow air-spaces there is not the same loss by unburnt fuel dropping through them. But on the other hand, even with a strong mechanically produced draught, narrow air-spaces have the disadvantage that they are more liable to get choked up with clinker.

It is important that the upper surface of the bars should be level; if not, the bars which project above the others will get burned. When replacing old bars it is generally advisable, when the grate is made up of two lengths, to place any new bars which may be required at the back end of the grate.

Clinkering.—All kinds of coal contain certain non-combustible constituents which in a greater or lesser degree fuse together and combine to form a slag on the surface of the grate, reducing thereby the rate of combustion owing to the air-spaces becoming choked. This gradually gets worse when the bars are old and worn and also when they get very hot. With old bars the slag runs down between the bars in the air-spaces more readily than with new bars and soon chokes them up. Various means have been proposed and tried to minimise this trouble which is always greatest with forced draught. In America the tops of the bars have been made concave in order that the hollows may get filled up with the clinker and the top edges only absorb heat directly from the fire. This, however, has not proved successful in producing any appreciable effect. The "Trident" firebars are cast with corrugations on the top running the full length of the bar. The hollows are filled with sand which, being a bad conductor of heat, keeps the bar underneath comparatively cool. This prevents the clinker from adhering fast to the bar and results in a longer life to the firebars and ease in cleaning the fires; considerable success has attended the use of this type of firebar in practice.

In order to keep the bars as cool as possible and thereby reduce this evil, they should be made fairly deep and the air-spaces should not be too narrow, which is again a matter for practical experience. Another method employed in practice to keep the bars cool and prevent the air-spaces from being choked up easily, especially when a high rate of combustion is required, is to flood the ash pit with water. For marine work, on forced-

draught trial trips it is often necessary to keep hose pipes playing sea water into the ash pits for this purpose.

In order to allow the firebars to expand lengthwise and therefore to minimise warping, one end is sloped off so as to slide up and down the dead-piate or the bridge-plate on which it rests. Two sets of bars are thus shown in Fig. 16, with the other end notched to drop over the intermediate cross-bars and so prevent the bars from shifting bodily.

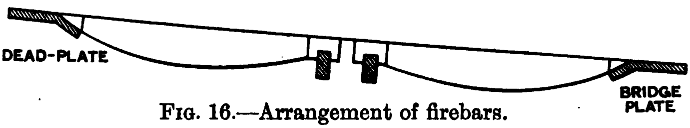


Fig. 16.—Arrangement of firebars.

4. Air Supply.—Although the greater part of the air required for complete combustion is admitted through the air-spaces between the bars, some air is also admitted above the fire through the furnace doors, but it is a matter of great difficulty to decide how much. The thicker the fire, the more air should be admitted through the doors as explained in Art. 5, Chap. II. If it were possible to work with very thin fires it would not be necessary to supply air through the doors, but unfortunately this is not practicable. With bituminous coal more air will be required above the fire than with Welsh coal, but in all cases the result aimed at should be to obtain complete combustion with the highest temperature obtainable before the gases reach the cool heating surface of the boiler; it is experience gained when burning any particular kind of fuel which decides the proper proportion of air to admit above the fire. In the case of the internally fired furnaces fitted in Lancashire boilers the length of the combustion chamber is so little that the flame is extinguished almost directly that it enters the furnace tube.

Another means of regulating the air supply to suit the rate of combustion required is afforded by the damper fitted in the main flue where the gases leave the boiler. When the damper is fully opened the maximum air supply is admitted; when completely closed, the air supply is shut off, this latter case being the normal condition when there is no load on the boiler and the fires are "banked."

5. Fire-Bridges.—A bridge is arranged at the back end of the firegrate in order to prevent the fire being pushed off, and also, by increasing the velocity at this point, to mix thoroughly the air and gaseous products with the object of completing combustion. It is built of firebrick and should not be too high; the higher the bridge, the less is the area of the flue at that point, and therefore the greater is the velocity of the gases. If the bridge is unduly high the velocity will be increased to such an extent that combustion will not be completed before the gases come into contact with the comparatively cool heating surface, the result being a reduced efficiency and larger deposits of soot in the flues as well as a smoky chimney. As already mentioned in Art. 1 (4), this is one of the greatest problems with which the designer has to contend, and since a very high bridge exaggerates the difficulty, the bridge should be kept as low as is practicable. In addition to the above, an excessively high bridge may so increase the velocity of the gases that the resulting "blow-pipe" action may cause undue wasting, and, with dirty feed-water, overheating of the furnace plates. In round figures it may be stated that the cross-sectional area of the space above the bridge should be about one-seventh of the area of the firegrate, but it should be remembered that it will depend upon the fuel used and also upon the rate of combustion.

Fig. 17 shows a common arrangement of furnace fitted to Lancashire boilers with natural chimney draught. The furnace tube A is riveted to the front end-plate of the boiler, the dead plate D, about 10 to 12 inches wide, being attached to the furnace tube and supporting one end of the fire bars. The other end of the front set of bars is carried on the cross bar E, which is also riveted to the furnace tube. The bridge B is of cast iron formed with a trough at the back into which the fire bricks are built. The bridge plate C is in front of the bridge and about 12 inches wide, the fire being pushed on to this plate when being cleaned; this arrangement greatly facilitates rapid and efficient cleaning of the fire.

6. Stoking and Smoke Prevention.—Given a well-designed furnace the amount of smoke produced can be greatly

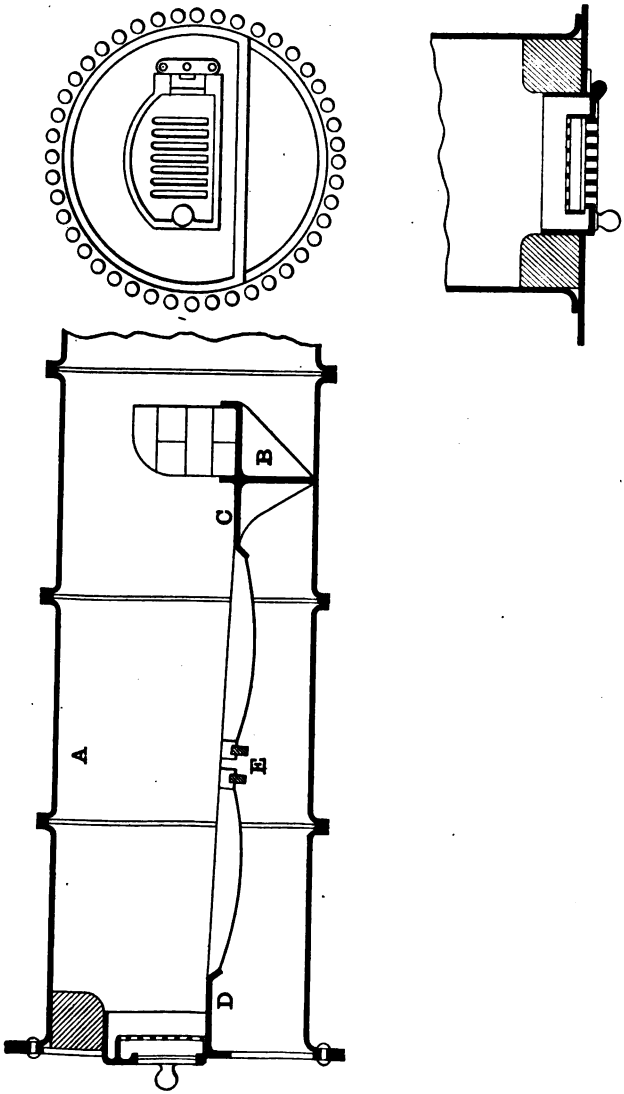


FIG. 17.—Lancashire boiler furnace.

reduced by careful attention to stoking, although with some varieties of bituminous coal it is impossible altogether to eliminate smoke in the ordinary type of furnace shown in Fig. 17. Good stoking is an art which can only be acquired after years of practical experience, and it is false economy for a steam user to employ unskilful stokers at low wages, because the result will almost certainly be an excessively high coal bill.

Mr. Stromeyer has shown "how easy it is to effect economies at the wrong end of the bill" by the following estimate of the cost of running a boiler for twelve months:—

	£
The coal bill for an average Lancashire boiler is about £300 to £600 per annum, say	500
The wages of a fireman who attends to several boilers may, per boiler, amount to, say . . .	25
Overhauling, repairs to furnaces and brickwork, cleaning of boiler, may cost	10
Boiler inspection, including incidental expenses, say about	2
Total	£537

From the above estimate it will be seen that about 90 per cent. of the total is for coal; any reduction in the remaining three items would have very little effect even if the coal bill remained the same; actually, the coal bill would be considerably increased and the extra cost of fuel would almost certainly more than counterbalance any saving, the net result being an increased cost of running.

He adds, "It is also very essential to compare the performances of the several stokers. Some men are incapable of efficient firing although they may be working themselves almost to death, whereas others, who appear to be working leisurely, may be exceptionally efficient, and it would be a very bad policy to dismiss such a man for a cheaper one, saving, say, £5 in wages, and paying, say, £100 in extra coal."

Under ordinary conditions of firing, smoke once produced is not burned (although by the supply of a large excess of air its density may be reduced), and if the conditions of combustion are such that smoke is produced the cure is effected either by

reducing the rate of firing or else by increasing the draught. It is frequently observed, that whereas in one works smoke is practically absent, at another works with the same coal and the same type of furnace and boiler the production of smoke is considerable, the only difference in the working in the two cases being in the rate of combustion. It should be remembered that whilst the production of smoke is an indication that the conditions of combustion are susceptible of improvement, the absence of visible smoke is not by any means an indication of proper combustion, for it may be brought about by too much air being supplied which dilutes the gases and renders the smoke less dense than it otherwise would be. Nor is the presence of smoke a sure indication that much waste takes place, for but a small quantity of carbon in the form of smoke will be sufficient to colour the flue gases from a boiler. Even a dense black smoke may only contain 1 per cent. of solid carbon, and if not accompanied by much CO, only implies a very small loss.

The object to be aimed at in good stoking is to keep the fire of uniform thickness all over the grate, and when firing is necessary, coal should only be thrown on the thin parts of the fire. If caking bituminous coal is used, the fire should be levelled at the same time as the caked pieces are broken up to clear the air-spaces between the firebars.

If Welsh coal is being used, the best results are obtained when the fuel is left undisturbed after it has once been thrown on the fire, and it is a test of good stoking when the fire can be maintained of uniform thickness for several hours without using a rake. In addition, it should be remembered that every time the furnace door is opened there is an inrush of cold air which chills the fire and heating surface, and therefore the doors should only be opened when necessary for firing. The thin places in the fire offer less resistance, and therefore more air passes through them, the result being that in these places combustion is fiercest, and there is the risk of excess of air passing through the fire when it is very thin. After firing up, these depressions will be filled up, but now, the other portions of the fire, being incandescent, will burn away quicker than the

places where the fire was thin before firing, since the original depressions are full of green coal. Great care is therefore necessary to keep the fire of uniform thickness. When there are several furnaces connected to the same chimney, the stoker who maintains the thickest fire will have the easiest work, and careful supervision is therefore necessary especially during forced trials on steam ships.

The furnace doors do not require to be opened so often with a thick fire, so that the periodic admission of cold air in rushes of comparatively large volume is thereby reduced. But on the other hand the temperature of ignition of the gases given off for example from a bituminous coal must be well maintained, and the air supply admitted through the furnace doors must be carefully adjusted so as not to chill the products of combustion. In the case of furnaces for water-tube boilers thicker fires can be used with advantage than in Lancashire boilers, on account of the great amount of heat stored in the firebrick lining, and the very much greater space for combustion.

The best thickness of fire to work with depends to a large extent on the fuel used, on the conditions of draught, and in practice partly on the personal opinion or even prejudices of the stoker. Speaking generally, the best thickness for internally fired furnaces as used in Lancashire and similar types of boilers is from 6 to 9 inches when working with natural chimney draught, and about 12 inches with forced draught. With externally fired boilers a thicker fire can be used with advantage for the reasons mentioned above.

Three methods of firing are in common use, the choice between them depending again upon the nature of the fuel, the draught, the load on the boiler, and the personal opinion of the stoker. One method of firing, known as the *spreading* method, is first to throw the coal evenly all over the back of the grate, and then, taking care to maintain the fire of uniform thickness, gradually to extend the layer of fresh coal towards the dead plate in the front until the whole surface of the fire is covered with a uniform layer of fresh coal.

In the *coking* method of firing the coal is thrown on the dead plate in the front of the grate; here the volatile hydro-

carbons are gently distilled off, and are carried by the air admitted through the furnace doors over the glowing fire where they are burned. When the evolution of the combustible gases has ceased, the resulting coke is pushed off the dead plate on to the firegrate where its combustion is completed. This is the best method to adopt with bituminous coal when the boiler is not overloaded, and the demand for steam is steady because the formation of smoke is reduced to a minimum. The disadvantages of this method consist in a tendency for the fire to burn unequally, and to develop thin places through which an excess of air may pass, and further, it is unsuitable if the boiler is being worked at its maximum power, or above its rated capacity, because of the difficulty in maintaining the necessary high rate of fuel consumption.

When there are several doors to the same furnace, another method is to fire through alternate doors, so that part of the fire is always in a state of incandescence, and burns the gases given off by the other part which is covered with fresh coal; it is a combination of the previous two methods. This method is fairly common since it enables full load to be maintained with a minimum amount of labour, and but little production of smoke.

A very large number of special devices have been designed and patented, having for their object the prevention of smoke when burning bituminous coal. In addition to the air supply through the fire and through the furnace doors above the fire, the split bridge shown in Fig. 18 is frequently used. Here a supplementary supply of air is admitted through the channel A in the bridge. By this arrangement smoke may be almost completely prevented and efficient combustion maintained, provided that the necessary care is taken to regulate properly the air supply so admitted. The door or damper B should only be opened by the stoker immediately after firing, and, when the evolution of gases ceases, be closed, otherwise an excess of air will be supplied which will lower the efficiency and only dilute the smoke instead of consuming it. A number of arrangements have been designed to secure the above results automatically; for example, the opening of the furnace door

for firing may be arranged to open the damper in the split bridge, and the latter is then closed automatically and slowly by a dash-pot mechanism, so that it is not closed for fully a minute. Or a sliding panel in the furnace door and the bridge damper are so coupled together that they are opened simultaneously by raising a weight; the fall of the weight is controlled by clockwork mechanism, which can be adjusted so that the extra air admitted above the fire and through the bridge is shut off at a predetermined time after firing.

In other cases hollow cast-iron firebars have been used in conjunction with a split bridge. In this arrangement the air is heated in its passage through the firebars and is delivered over the fire. A further advantage which is claimed is that the

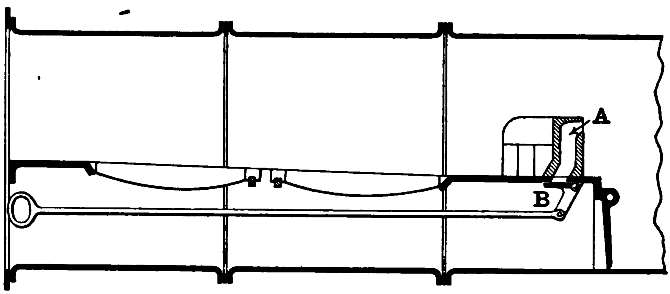


FIG. 18.—Split bridge.

bars are kept cool, and so the evils of clinkering are reduced, but it is rather doubtful how far this claim is justified. For this reason plain bars are generally used in practice.

When designing a boiler furnace for any particular coal reliance should be placed on the air supply through the air-spaces between the firebars, and through the furnace doors in order to obtain proper combustion with a minimum of smoke. It is when coal is used of a more bituminous nature than the coal for which a furnace has been designed that one or other of the above devices may be tried with advantage.

7. Furnaces for Water-Tube Boilers.—The most efficient portion of the heating surface of an internally fired boiler consists in the furnace plates which are directly exposed to the

radiation from the fire, the rate of heat transmission being much greater there than in any other part of the boiler (Art. 13, Chap. V.). This results in a tendency for the furnace temperature to be reduced below the high value necessary for complete combustion. As already mentioned in Art. 2, the area of the firegrate depends upon the diameter of the furnace tube, but in the case of a water-tube boiler furnace there is no such restriction in either the grate area or the volume of the combustion chamber, and further, it is possible to maintain a high furnace temperature, without straining the boiler, because the furnace is not surrounded by a comparatively cool metal wall.

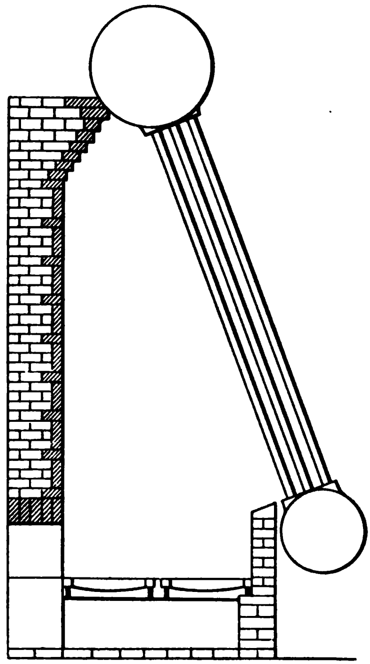


FIG. 19.—Furnace for "Woodeson" water-tube boilers.

By employing an external brick furnace it is very easy to combine a large furnace capacity with a high temperature of combustion, but as a set-off to this advantage, the radiation loss will be greater than from an internal furnace. In the case of water-tube boilers the water tubes should not be placed too near the fire; otherwise the flame will be extinguished before combustion is complete (Art. 1). Ample space should therefore be allowed between the firegrate and tubes for combustion as illustrated in Fig. 19, which shows the furnace used in the "Woodeson" water-tube boiler. It is also much easier to obtain smokeless combustion of cheap inferior coal in a water-tube boiler furnace than in the furnace of a Lancashire or

similar type of boiler on account of the large combustion chamber in the former.

8. External Furnaces for Lancashire Boilers.—The tubes of the Lancashire boilers are too small to allow of their being themselves lined with firebrick without unduly reducing the grate area and combustion space, and this would also render the most effective portion of the heating surface, viz. the furnace plates (Art. 7), almost useless. Hence, in order to obtain the same result and render possible the smokeless combustion of bituminous coal in the Lancashire type of boiler, external brickwork furnaces are frequently used in districts where coal is expensive. In this case the furnace tubes are carried straight through from end to end of the boiler, and a brickwork furnace is built in front of the boiler, the furnace gases passing through the tubes, then underneath and round the sides of the boiler shell to the chimney in the usual way. Fig. 20 shows a battery of two Galloway boilers fitted with an external furnace. (Cp. also Fig. 32).

9. Furnaces for burning Anthracite. — Although anthracite is chiefly used in gas producers, it is occasionally used in special boiler furnaces for raising steam. Pure anthracite burns fiercely with a very short flame, giving off very little volatile gas. A large combustion chamber is therefore unnecessary, but a special firegrate is required. In the usual form of anthracite furnace firebars are dispensed with, the surface of the grate being constructed of flat plates having conical projections pressed in them, each projection being perforated with a hole through which air is admitted beneath the fire. Although this fuel does not give off any volatile hydrocarbons when thrown on to the fire, it is found necessary to admit an auxiliary supply of air above the fire through the furnace doors in order to burn the carbon monoxide (CO) produced, particularly when a thick fire is being worked (Art. 5, Chap. II.).

10. Furnace for burning Bagasse and Refuse Fuel.—A considerable number of furnaces have been designed to burn this fuel, and the variations in the details are largely

due to local working conditions and the character of the sugar cane from which the bagasse is obtained. Fig. 21 shows a furnace designed to burn this fuel with forced draught and fitted to a Babcock and Wilcox water-tube boiler. The furnace

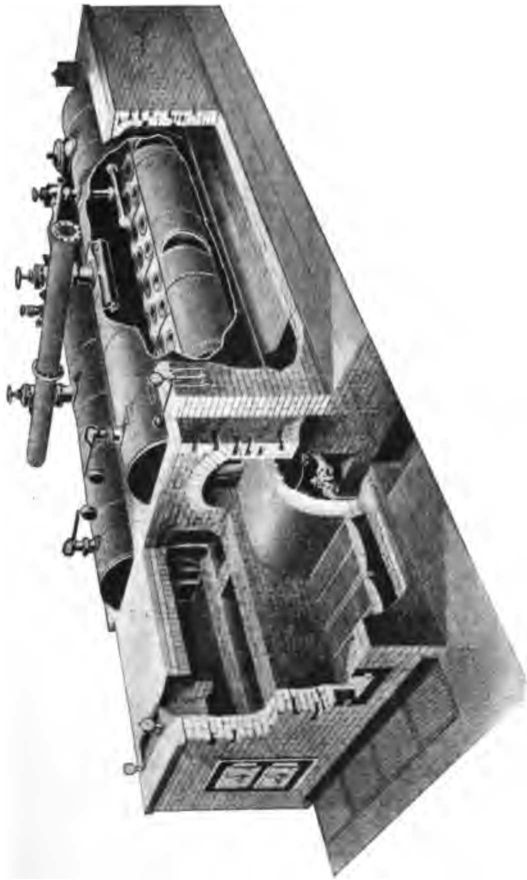


FIG. 20.—Galloway boilers, with external furnace.

is external to the boiler and is lined with firebrick, the fuel being admitted through the roof and air under pressure blown into the fire through the holes shown.

Another type of furnace for use with natural chimney

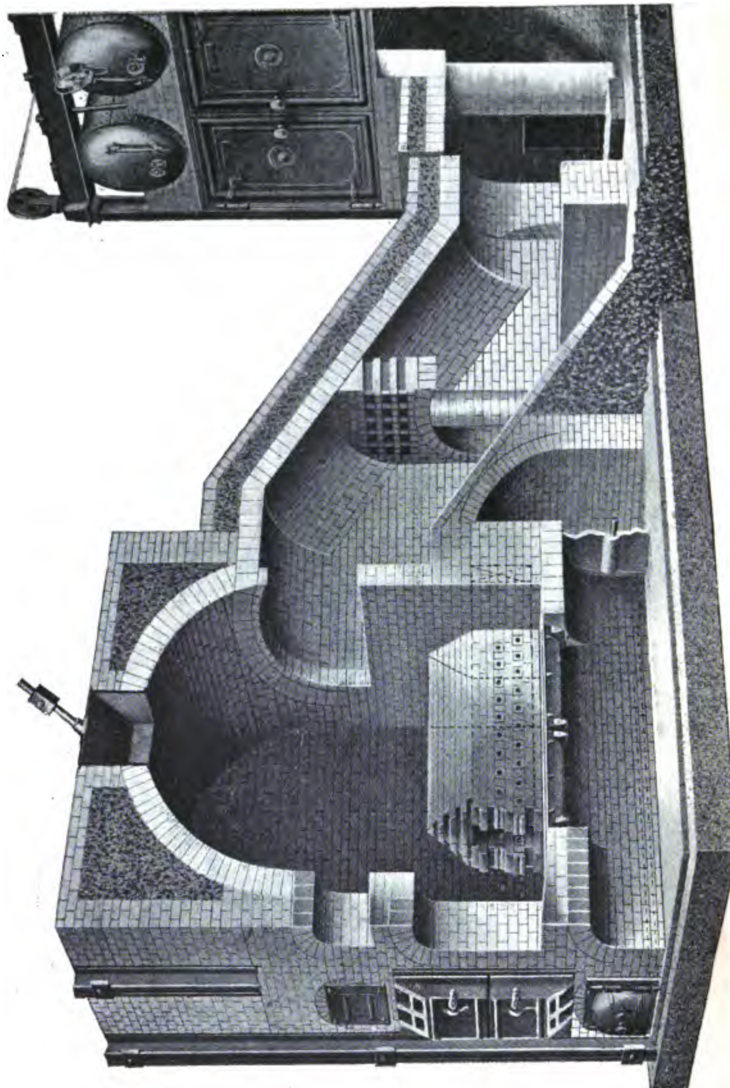


FIG. 21.—Babcock & Wilcox cold blast forced draught bagasse furnace.

draught is shown in Fig. 22. It will be seen that the fuel is fed into the furnace from a firing door placed above the grate, down a tube fitted with a self-closing door and falls on to a stepped grate. As the fuel is burned it falls down to the bottom of the furnace, and is replaced by other fuel which has been previously dried by the surrounding heat.

Many difficulties are met with when trying to burn low-grade fuels such as lignite, peat, and the like, containing a high percentage of moisture, ashes, etc. These difficulties arise from their poor calorific value, from the cooling effect due to the moisture developing large quantities of steam and thus causing the temperature to fall below the minimum required to keep up combustion, and from the large quantity of slag which must be removed from the furnace. Such fuels have been successfully burned in the step grate furnace (Fig. 22), but usually only with small or medium-sized boilers and furnaces of relatively large dimensions, the firing and removal of the slag being done by hand. Since in many cases very large boilers are now used and installations having a large number of such boilers are employed, the furnaces that would be required of this type would be of almost prohibitive size.

11. Combination of Step-grate and Chain-grate.—In order to overcome the above difficulties the furnace shown in Fig. 23 has been designed.* The furnace is a combination of a chain-grate A with an arrangement specially devised for gradually drying and igniting the fuel, and consisting of a chamber B fitted with a step-grate C. The steps D of the grate are lined with fireproof material in order to promote and complete the reverberating action of a fireproof arch F built above the step-grate and forming the top of the chamber B. Such a fireproof arch alone would not be capable of maintaining the necessary temperature. Its action is assisted by the fireproof lining of the steps, which, by its direct action as well as by its preheating of the air passing through the spaces between the steps, has, it is claimed, enabled fuels containing up to 60 per cent. of moisture to be burned satisfactorily, which would otherwise have been impossible.

* See *The Mechanical Engineer*, June 17, 1910.

The reverberating action ensures gradual penetration by heat of a rather thick bed of fuel, the fuel being dried up as it

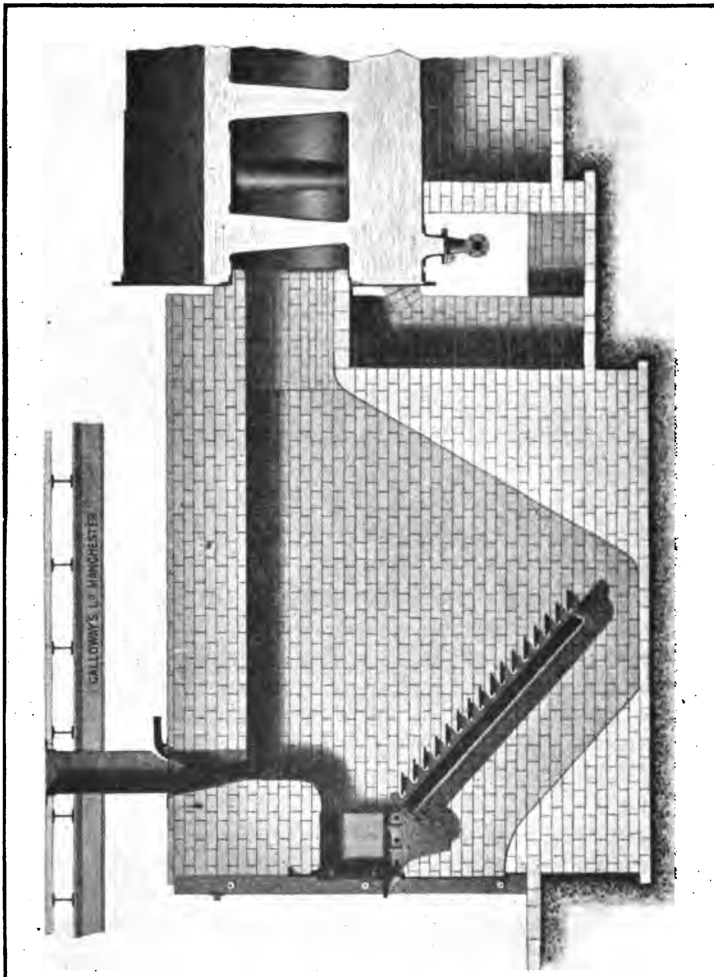


FIG. 22.- Furnace for burning bagasse in Galloway boiler.

descends the steps. The number of steps provided is such as just to ensure the complete drying of the fuel and its delivery

well ignited at the front of the chain-grate, which carries it backwards until it is completely consumed. Owing to the fuel being surrounded by fireproof walls on all sides and being exposed to air which becomes gradually hotter as the fuel descends the step-grate, the moisture evaporates slowly and not suddenly, so that the tendency to cooling is almost prevented. The arch deflects the water vapour downwards over the fire, where it is dissociated and mingles with the flames as water gas.

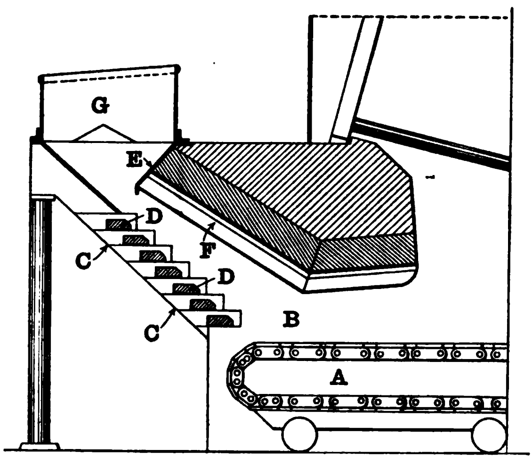


FIG. 23.—Boiler furnace for low grade fuels.

When a step-grate is combined with a chain-grate the adjustment of the thickness of the fuel bed presents some difficulty. Owing to friction against the adjacent walls, the motion of the fuel is retarded at the sides of the step-grate as well as of the chain-grate; the sides, being thus less supplied with fuel and subjected to a more intense reverberatory action, show a tendency to develop a more active combustion and finally end by becoming bare, when an excess of air will pass through. In the middle portion of the grate, however, the fuel accumulates and the air available for combustion there is less. To obviate this, in the front portion B (Fig. 23) of the grate is fitted a

regulating plate E, which exercises its action at the point where the fuel, issuing from the hopper G, reaches the first step of the step-grate. The regulating edge of this plate E is either curved, or the corners at the two ends are cut away. The effect of this shape of the regulating plate is that the thickness of the fuel along the sides of the grate is increased. The bottom step, or several of the last steps at the bottom of the step-grate, have their edges curved in a similar way, or else have the corners cut off at the two ends. In this way the inclination at the bottom of the step-grate, and therefore the speed of descent of the fuel, is increased on either side of the grate. These two arrangements conjointly overcome the tendency of the fuel to accumulate in the centre portion of the grate, and ensure the maintenance of a regular distribution throughout.

12. Mechanical Stokers.—It is frequently stated that the use of mechanical stokers affords a simple and effective solution of the smoke problem. While this may be true in many cases, it is an undisputed fact that their use does not always prevent the formation of smoke. It has been shown in Art. 6 above, that the amount of smoke produced depends upon the rate of combustion as well as upon the type of coal used. A cheap, inferior, bituminous coal which could not be effectively burned by hand firing, may frequently be used in conjunction with mechanical stokers and satisfactory results obtained.

The utility of a mechanical stoker lies primarily in obtaining at regular intervals the mechanical feeding of such small quantities of coal that the time taken between two charges suffices for the gases to be completely burned; secondly, the avoidance of loss due to the admission of excess of air through opening the furnace doors when firing by hand; and thirdly, the reduction in human labour, one man being able to control a greater number of furnaces than is possible when hand firing is used. With coal of fairly good quality the efficiency attained with skilful hand firing is quite as high as when mechanical stokers are used. Apart from the possibility of being able to burn cheap and inferior bituminous coal, the saving which results from the

use of mechanical stokers is entirely due to the reduction in labour, particularly in large plants where mechanical conveyors are used for feeding the coal to the hoppers of the various stokers.

A large number of different types of mechanical stokers are in use, each with its own particular advantages; lack of space prevents the description of more than one or two types.

Sprinkling Stokers.—With this type of stoker the *spreading* method of firing, described in Art. 6 above, is carried out by mechanical means.

Coking Stokers.—With this type the *coking* method, described above, is mechanically maintained. Both sprinkling and some forms of coking stokers are suitable for use in internally and also in externally fired furnaces.

Underfeed Stokers.—This type of stoker is usually adopted with externally fired furnaces, and requires a coal of good quality which contains a minimum amount of ash and which produces very little clinker. The method of firing is entirely different from the two methods described above, the fresh coal being fed into the furnace from underneath. The gases force their way up through the incandescent fire and are there burnt.

13. Chain-Grate Stoker.—This stoker, which is of the coking type, is only used for externally fired boilers on account of the large area of the fire grate. It consists of an endless chain or belt constructed of a large number of short cast-iron bars pinned together and passing over a sprocket wheel at each end. The front wheel is driven by mechanical means, causing the upper part of the chain to travel with uniform speed from front to back carrying the coal with it, the speed being such that the coal is completely burnt when it reaches the back end. The speed is capable of adjustment to meet the load on the boiler and therefore the rate of combustion required, and a detachable handle is usually fitted so that the grate may be rapidly wound in by hand to meet a sudden demand on the boiler. The coal is fed on to the front of the grate, and the thickness of the bed

of fresh coal is regulated by a "trimming" plate which can be raised or lowered by hand. It is then left undisturbed until it is entirely consumed. The grate is mounted on a carriage and wheels so that it can be withdrawn away from the boiler, Plate I. shows a chain-grate stoker applied to a Babcock and Wilcox boiler.

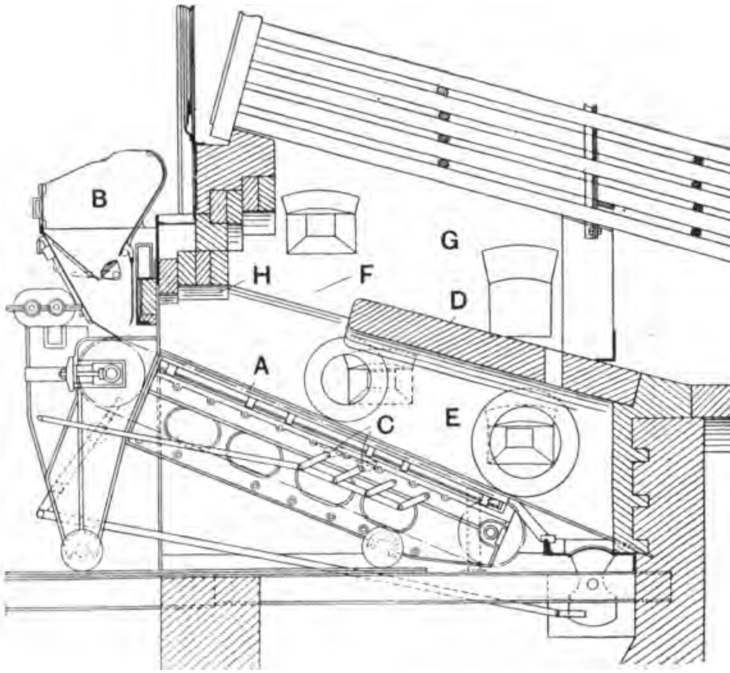


Fig. 24.—New type of Babcock & Wilcox furnace, with chain-grate stoker.

A new type of chain-grate stoker applied to a water-tube boiler has recently been introduced by Messrs. Babcock and Wilcox, and is shown in Fig. 24. A chain-grate stoker arranged at an inclination to the horizontal is employed in conjunction with an inclined rear arch and a front arch, both of which are built of firebrick. The furnace consists of an automatic stoker having a uniformly inclined chain grate A for carrying the fuel

which is fed into the furnace through the usual hopper B in front. The stoker is provided with a series of air ducts C under the grate which are operated by levers and rods from the boiler front, and over the rear end of the grate is built a firebrick arch D extending forwards to a point about midway of the length of the grate. A short arch is also formed at the front over the forward end of the grate rather closer to the grate surface. A comparatively large reverberatory chamber E is thus formed over the rear portion of the furnace in which the fuel is burned by the regulated inflow of air through the vanes under the grate. The flame from this rear chamber is projected forward in its passage towards and through the opening F between the front and rear arches, and coming in contact with the comparatively green fuel on the front part of the grate ensures its proper combustion. The hot gases passing through the opening F circulate within a secondary combustion chamber G below the water tubes and thence pass around these tubes, being guided by the usual baffles on their way to the chimney outlet.

14. Bennis Sprinkling Stoker.—This stoker, suitable for use in a Lancashire boiler, is shown in Fig. 25. Under each hopper (there being two hoppers of about three cwt. capacity to each Lancashire boiler) is a cast-iron feeding box, in the interior of which is a simple pusher plate C with an adjustable reciprocating motion. The fuel falls in front of the pusher plate and is pushed, by its movement, over a ledge formed by the bottom of the feeding-box. The weight of fuel so pushed over is regulated by means of an adjustable cam E on the driving shaft, so that the rate of feed can be seen by noting the position of the cam. By moving the screw G the lever F adjusts the position of the cam E in order to give by the bent lever P the necessary stroke to the pushing plate for the rate of feed desired. The fuel thus pushed over falls into a "shovel box" from which it is projected into the fire at intervals by an angular shovel H; by this means it is effectually scattered over different portions of the grate. The shovel H has a striker J attached to its free end and swings about the pin O, being actuated as follows:—The end of the shovel is attached by means of the rod K to a piston L which is propelled backwards by means of a helical spring M enclosed in

a cylinder as shown. The spring is brought to rest by means of an air cushion and all shock and jar on the boiler front are thus prevented and noiseless operation results. The cam N, which

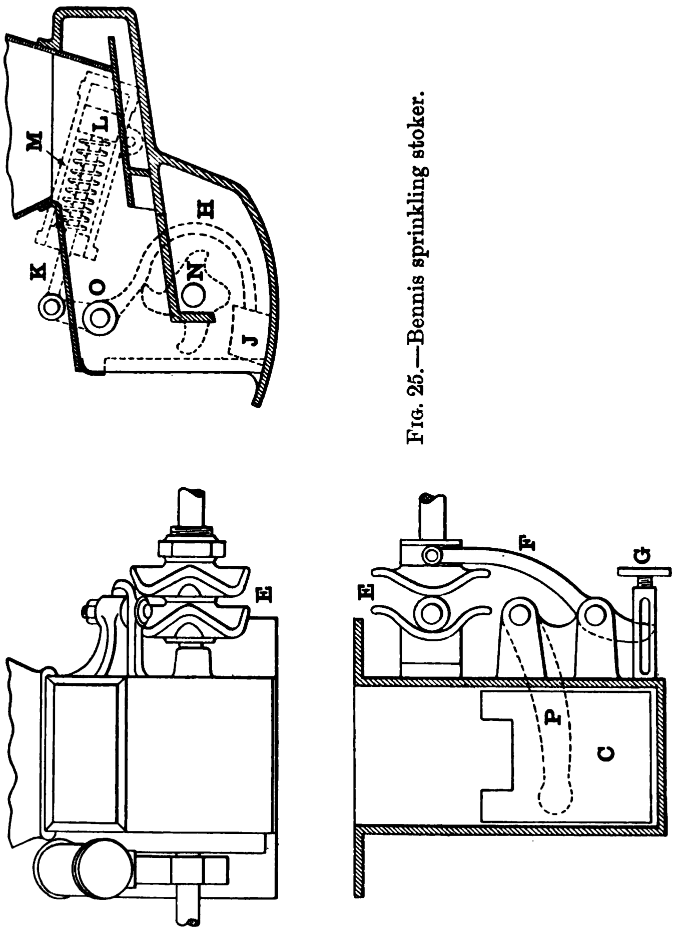


Fig. 25.—Bennis sprinkling stoker.

draws back the shovel and in so doing compresses the spring M, has four different lifts; the effect of this is the scattering of the fuel in four divisions of the grate, each about 18 inches long, so

that in a furnace grate 6 feet long, the fuel is thrown on only a quarter of the fire at once and each portion of the fire has time to become incandescent between the charges.

When using low class or waste fuels that contain a large proportion of ash and clinker, the air-spaces between firebars of the ordinary type soon become choked up, and in order to remove the clinker automatically special firebars are used in this furnace. A series of tubular troughs extend the whole length of the grate, being placed close together and protected from direct contact with the fire by being covered with short interlocking grate bars A (Fig. 26), about 2 feet in length. The back of the tubular fire trough is protected by a solid block B, and in the event of any of these interlocking grate bars or blocks being damaged they can be replaced without interfering with the rest of the grate. Air is forced through the bars by means of fine jets of highly superheated steam which blow the air into the tubular bars at a fairly high pressure and thus permit of a high rate of combustion being used. The cross section

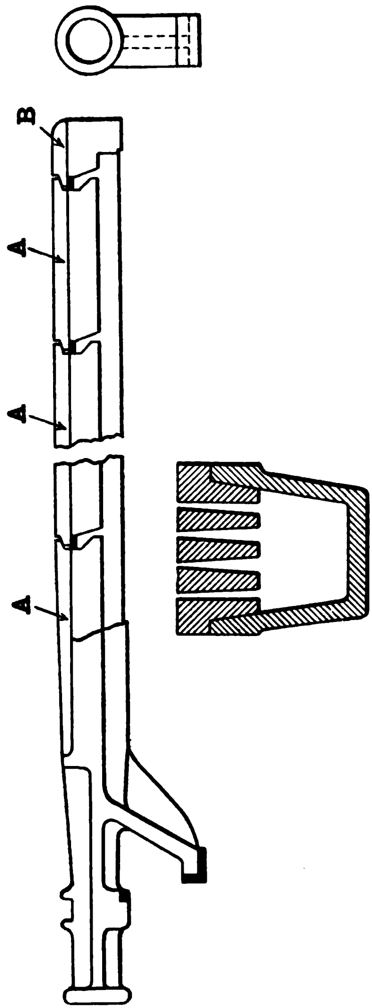


Fig. 26.—Bennis patent compressed air bar.

of a trough with its bars is shown to a larger scale underneath.

The troughs are all moved into the fire together for a distance of about 2 inches and are then drawn back by means of cams on a transverse shaft (not shown). The clinker and ash are slowly carried along by this action to the end of the bars, from which they drop over into a closed chamber which is cleaned out about once or twice every day.

15. Bennis Coking Stoker.—The general arrangement of the latest type of this stoker is shown in Fig. 27. Underneath the hoppers, duplicate feed-boxes are arranged which feed the fuel into the furnace alternately and intermittently; the method of regulating the feed is the same as in the Sprinkling Stoker (Art. 14). Below the feed-boxes a large fire door of considerable width is provided which prevents any air entering the furnace at the front. Below the fire door a series of dead-plate bars A is provided, constituting a live dead-plate designed to admit air under pressure so as to cause the rapid lighting up of the fuel. Below these bars the main furnace consisting of the self-cleaning tubular bars described in Art. 14 is situated, the bars moving to and fro. The two outside troughs, however, have only a very small movement, so that the fuel on them does not travel as rapidly from the front as in the rest of the furnace. The effect of this combination is that there is always a white-hot fire near the front at the two sides, and the blast of air through the narrow air-spaces between the bars, and of the live dead-plate bars, is such that the whole of the fuel fed across the fire is ignited within a few inches of the live dead-plate bars at such a speed that the machine will burn nearly as much fuel per hour as the sprinkling stoker.

When the partially coked fuel leaves the live dead-plate it falls several inches on to the main grate, and in so doing is broken up into smaller pieces, thereby allowing a free play of air through the incandescent coke. The result is that the firebed is rendered quite porous instead of remaining a semi-solid mass as it does in the ordinary coking stoker. The fuel travels backwards along the self-cleaning bars, meeting, as it passes, varied

pressures of air, proportioned to the thickness of the fire on the grate at each point.

The rate of combustion is controlled, and the necessary

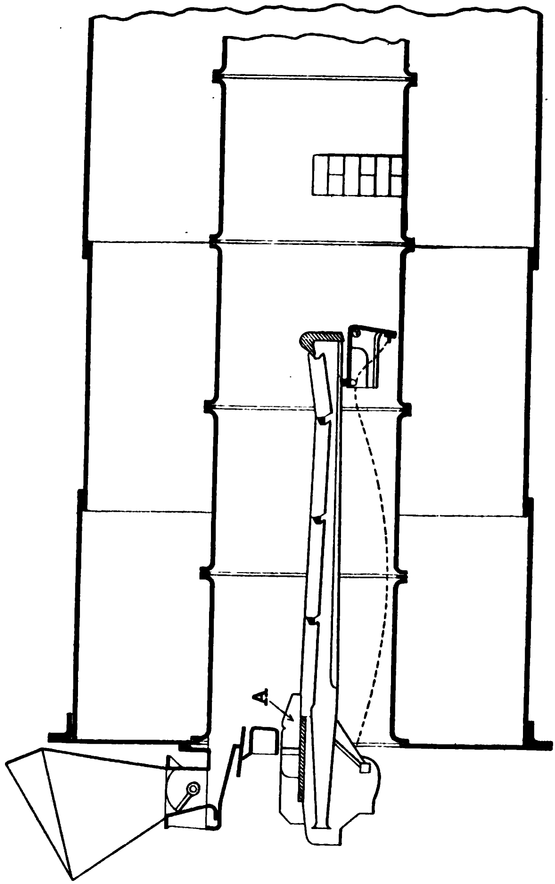


Fig. 27.—Bennis coking stoker.

degree of flexibility obtained by varying the air supply to the compressed air furnace troughs and by the adjustment of the damper of the boiler.

16. Advantages and Disadvantages of Mechanical

Stokers.—Although in some cases, mechanical stoking possesses several advantages over hand-firing (see Art. 12) there are in some instances certain disadvantages which may result from its use, and in spite of the many ingenious improvements which have been made with mechanical stokers they are still inferior to hand stokers when dealing with a sudden fluctuation in the load on a boiler. In the case of a coking stoker, for instance, there will be for any suitable fuel and load on the boiler, a certain rate of working and of air supply combined with thickness of fire which will give the most economical results. If now the load on the boiler suddenly increases or decreases it will take an appreciable interval of time before the necessary economical conditions are again attained, *i.e.* the draught, thickness of the fuel bed, and the speed of the stoker will all have to be readjusted. Such changes in load demand similar changes in stoking to which men easily get accustomed, but the best mechanical stokers have not yet attained the elasticity of hand-firing.

A change in the qualities of the coal may also in some cases give serious trouble. A stoker designed and regulated for burning smokelessly cheap slack will invariably prove unsuitable for use with a better class of coal. Mr. Stromeyer * mentions a case in which a stoker was specially fitted to burn cheap slack, and it worked very well; for some unascertained reason, however, Welsh nuts were fed into the furnace, presumably by mistake, with the result that the cast-iron parts fused together and had to be chipped out, an operation which lasted two weeks. It should be remembered that a mechanical stoker is compelled to work away irresistibly against all sorts of obstacles, and may ultimately tear itself to pieces. This accounts for the varied results obtained with mechanical stokers, some working smoothly for years, whilst with others the cost of repairs and increased depreciation, far outbalances any saving in fuel consumption and labour which they may effect.

Large water-tube boilers are always fitted with mechanical stokers (see Table I., p. 7) on account of the saving in labour

* *Memorandum to the Manchester Steam Users' Association*, 1909.

and the more uniform furnace conditions obtained. The furnace, lined with firebrick, has a large grate area and a combustion chamber of ample capacity which allows a thick fire to be used and a thorough mixing of the gases giving complete combustion before the heating surface is reached, without the ill effects resulting from the inrush of cold air during the frequent opening of the furnace doors rendered necessary with hand-firing.

On a hand-fired grate it will often happen that at some point the thickness of the bed of fuel is so well suited to the draught that locally a very intense flame impinges against the furnace plate, but this will be of short duration only and before any damage results to the plate the thickness of the fire at that point is reduced, and it may be a long time before a similar condition recurs; when it does, this condition in all probability will be obtained at another point. With a mechanical stoker, however, if an intense flame is once produced at any point on the grate it will tend to continue at that place because the thickness of the fire and the supply of air are steadily maintained by mechanical means.* If this persistent local heating be sufficiently intense it may lead to a softening of the furnace plate and slow bulging, or, in the case of a water-tube boiler, to blistering and burnt out tubes.

17. Furnaces for Burning Oil Fuel.—The use of oil fuel for steam-raising purposes has many advantages over coal, and in countries where crude petroleum is plentiful, *i.e.* in Russia and America, it is extensively used. Hitherto, the greatest drawbacks in this country and in other places at a great distance from the oil-fields, have been the *cost* of liquid fuel compared with that of coal and the *uncertainty* of obtaining a steady and sufficient supply. In the neighbourhood of the Caspian Sea "astatki," or heavy petroleum refuse, is plentiful and about half the price of coal, whilst its calorific value is about one and a half times that of coal. It is largely used for fuel in the boilers of steamers on the Caspian, and in locomotives in the south-east of Russia. Oil-burning furnaces are also well adapted for water-tube steam boilers as a stand-by in electric

* Mr. C. E. Stromeyer, *loc. cit.*

lighting central stations, owing to the rapidity with which steam can be raised. Further, the rate of combustion can be regulated to meet, with ready response, the sudden changes in demand for steam due to the variation of power required for fog or maximum output of electric current.

The advantages of liquid fuel over coal may be briefly summed up as follows :—*

(*a*) The calorific value is about 30 per cent. higher than that of high-grade coal, so that the weight of oil is less than the weight of the coal to give the same evaporation.

(*b*) The space required for storage is less than that required for an equal weight of coal. About 40 per cent. more heat can be stored in the same volume and at a greater distance from the boilers without extra expense. For marine work this is a great consideration.†

(*c*) The facility in taking in a fresh supply of liquid fuel aboard steamships and the absence of coal dust and grit in "coaling."

(*d*) Oil does not deteriorate by storage, but maintains its heating value indefinitely in ordinary ventilated storage tanks.

(*e*) Lower temperature in the boiler room.

(*f*) Less heat is lost up the chimney owing to the cleaner condition of the boiler-heating surface, and to the smaller amount of air which has to be supplied for a given calorific capacity of fuel.

(*g*) Higher efficiency due to (1) more perfect combustion with less excess air; (2) more equal distribution of heat in the combustion chamber as the furnace doors do not require opening; and (3) the small amount of soot which is deposited on the heating surface.

(*h*) Ease with which the fire can be regulated from a low to a most intense heat in a short time or entirely extinguished instantly in cases of emergency, such as the water level falling out of sight in the gauge glass, and quickly relighted when the emergency is over.

* Based on a paper on "Oil Fuel for Steam Boilers," by R. R. T. Collins, *Journal of the Am. Soc. Mech. E.*, August, 1911.

† 1 ton of oil occupies about 38, and 1 ton of coal about 44 cubic feet.

(*i*) Smoke can be entirely eliminated with a properly designed furnace.

(*j*) No cleaning of fires is necessary, and the boilers can maintain their maximum evaporation continuously if required.

(*k*) Much lower cost for handling oil fuel, as it runs by gravity or is pumped into and out of storage to the boilers.

(*l*) Absence of coal dust and ashes, so that everything in the boiler-room can be kept clean. There is no expense for handling and removing ashes, no firing tools are required, and no clinkers require removing from the bars or furnace side walls. The laborious work of stoking, cleaning fires and coal trimming is therefore done away with, and considerable saving is effected in the cost of labour.

The disadvantages of oil fuel are :—

(*a*) Its low flash point. Oil fuel should have a flash point not lower than 140° F., and with such oil, handled by men of ordinary intelligence and common-sense, there is practically no more danger of fire or of gas explosions than with coal. The pipe system must be most carefully constructed and maintained free from the least sign of leakage.

(*b*) The ordinary insurance or city requirements specify that storage tanks for oil fuel shall be placed underground and at least 30 feet from the nearest building. In the case of a plant in the congested districts of a city this is likely to be prohibitive.

(*c*) With boilers using feed water of considerable scale-forming properties, the cost of repairs is likely to be increased by changing from coal to oil, owing to the very high temperature developed in the furnace. With a proper setting for burning oil, however, the repairs due to overheated tubes or surfaces should be less, or at any rate not greater than, with coal, unless the feed water is very bad.

Requirements for Complete Combustion of Liquid Fuel.—In order to obtain complete combustion, the oil must be reduced to a fine spray, or, in other words, completely "atomised"; by so doing, each particle of oil is brought into contact with the proper amount of air. The mixture of oil

spray and air thus produced should be burned in a furnace lined with a refractory material and of sufficient capacity to afford complete combustion before the gases come in contact with the boiler-heating surface.

The first condition is fulfilled by selecting a proper burner, and the remaining conditions can generally be obtained by making slight changes in, or additions to, existing furnaces. The question whether to use steam or air for atomising the oil seems generally to have been decided in favour of steam, for experimental results show that it takes about the same amount of steam to operate the air compressor as it does to atomise the oil at the burner, and the additional investment and complication involved, with greater possibility of interrupted service, are avoided. On the other hand, the use of steam results in a higher specific heat of the flue gases leaving the boiler and hence in more heat being lost than if air were used (see Art. 9, Chap. II.). There are also mechanical atomisers which deliver the oil through a special nozzle under a pressure of from 40 to 150 pounds per square inch, but generally speaking this type has not proved satisfactory in practice.

"Heating of the oil is an aid to economical combustion, and should take place as near the furnace as possible and be carried as high as safety permits, but not so high as to cause the oil to decompose and carbon to be deposited in the supply pipes. If preliminary heating is limited to the temperature of the flash point of the oil used, there can be no trouble from the above-mentioned causes.

"In oil burning, although a certain amount of skill is required for hand adjustment of the burners to obtain the best results, still, with automatic regulation, the skill is reduced to a minimum, the principal work of the fireman being to see that the oil pump is kept in constant operation and that the burners do not become clogged with small particles of foreign matter, scale, etc., especially when the installation is new. Strainers of proper design, however, introduced on the suction line to the pump and also between the pump and the burner, will reduce this trouble to a minimum. Burners should be so installed that they can be easily disconnected from the piping

and taken from the furnace for the removal of any foreign substance from their restricted orifices.

"One of the most important questions in the combustion of liquid fuel is the regulation of the air supply in such a way as to obtain perfect combustion before the gases come in contact with the heating surfaces of the boiler. This can be done with an automatic damper regulator, although its adjustment is rather difficult. It is therefore usually accomplished by hand regulation of the damper when considerable variations in the load take place. This is supplemented by changing the position of the ashpit doors, which are kept partly closed until a slight tendency to make smoke is noticed in the furnace, when they are opened until this tendency disappears; or, better, by using an Orsat or continuous CO_2 gas analyser to determine the position of damper and ashpit doors which gives most complete combustion under certain constantly recurring conditions."*

A very large number of oil burners or atomisers have been patented from time to time, a description of which is beyond the scope of this book. The important features which should be embodied in all burners are: a construction that will allow for quick inspection, easy removal of all foreign matter which may clog the burner at any point, rapid and cheap renewal of any parts which are subject to wear, combined with an easy method of installation.

In spite of the various principles involved in burner construction, the success of an oil fuel installation depends not so much on the type of burner or atomiser used, as on the method of its installation, and the intelligence with which it is operated after the installation is made. The work of designing and constructing such installations should be entrusted only to those having extended practical experience in burning oil fuel, and the operation of these installations should not be entrusted to unskilled labour, since in order to obtain the economic results possible from the use of oil fuel, it will be necessary to employ men of intelligence and skill as firemen.

* From the above-cited paper by R. R. T. Collins.

Holden's System.—The method of burning liquid fuel invented by Mr. James Holden on the Great Eastern Railway, and now extensively adopted on many railways abroad, as well as in numerous stationary and marine boilers, is to have the furnace arranged for either coal or oil fuel, depending on their price so that either oil alone may be burned, or coal and oil in convenient proportions, or coal alone. Mr. Holden used a mixture of tar and oil refuse from gasworks in conjunction with coal. The furnace or firebox of the boiler is lined inside with firebrick to protect the plates, and to act as an accumulator and reservoir of heat to keep up a constant high temperature in order that combustion may be complete before the gases reach the cool heating surface of the boiler.

Fig. 28 shows a longitudinal section of the latest type of Holden's burner.* The controlling valve A regulates the inlet of liquid fuel to the annular space B within the body of the apparatus; C is the steam injector nozzle, D the air inlet nozzle open at its rear end for the admission of air, and E is the port through which steam is admitted to the annular space F surrounding the air inlet nozzle D, and to the interior of the steam injector nozzle C. Steam is supplied through the pipe G to the interior of the injector nozzle C, a suitable controlling valve (not shown) being provided for controlling the admission of steam. H indicates the liquid fuel delivery nozzle into and through the chamber J, in which the liquid fuel is forced by and together with the steam issuing from the inner end of the injector nozzle C, the liquid fuel and steam thence issuing in jets through the outlet orifices in the front end of the nozzle.

The construction of the delivery nozzle H is such that its chamber J is eccentric in relation to the axis or longitudinal centre line of the steam injector nozzle C, and of the annular space or chamber B into and through which the liquid fuel is drawn by the action of the steam injector nozzle C. The construction is such that the bottom of the chamber J is nearer to the axis or longitudinal centre line of the steam injector nozzle C than is the top of the chamber J. K, M and N are pairs of liquid fuel outlet orifices in the front end of the nozzle H,

* From *The Mechanical Engineer*, July 14, 1911.

the orifices of each pair being arranged at an angle to each other, so that the two apertures that constitute a pair converge

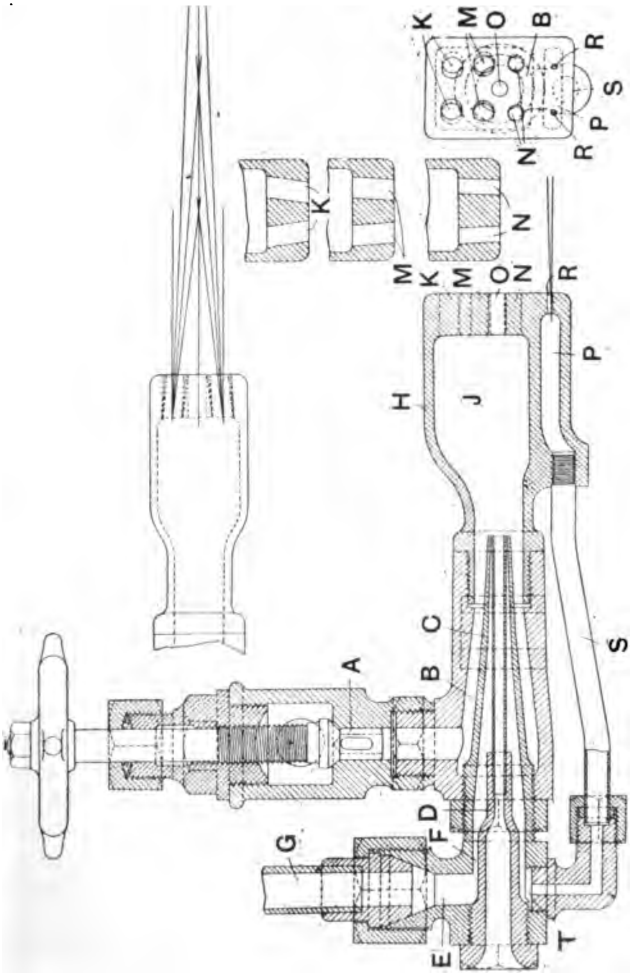


FIG. 28.—Holden's liquid fuel burner.

or slope towards each other from their inner to their outer ends. As will be seen from the sectional plan views, the apertures M

slope towards one another at a more acute angle than do the apertures K, and the apertures N slope towards one another at a more acute angle than do the apertures M. By this means the jets of liquid fuel issuing from the respective pairs of outlet apertures, meet or strike one another at different distances within the firebox.

The lowest apertures N are situated at only a slight distance above the bottom of the chamber J, and are of smaller diameter than the apertures K and M. The intermediate aperture O is arranged centrally between M and N, while P indicates a steam chamber at the lowest part of the nozzle H, R being two small steam jet orifices leading out of the steam chamber and sloping from their inner ends towards the centre of the nozzle H. The steam pipe S is connected at one end with the rear end of the steam chamber P, and at the other end with the part T, which is provided in the wall of the injector nozzle C, and is in communication with the annular space F, which space forms a continuation of the interior of the injector nozzle C.

Should, from any cause, oil drop down from the lowermost or from any of the liquid fuel outlet orifices of the nozzle, such drops of oil, instead of dropping into the furnace in an unatomised condition, will be atomised and carried into the furnace by the jets of steam issuing from the small steam jet orifices R. The steam in the chamber P of the nozzle H also serves to maintain the liquid fuel within the main chamber J at a temperature above that at which it enters the apparatus for the purpose of increasing the fluidity of the liquid fuel.

The firebox (Fig. 29) of the boiler is protected by a bridge and lining of firebrick towards which the spray is directed, and the firebars are covered with a thin layer of incandescent coal and cinders mixed with broken brick, which prevents the inrush of too much cold air, and keeps up the temperature high enough for complete combustion. The loss of heat owing to inrushes of cold air which occur when stoking coal fires is avoided, and combustion is complete without any appreciable amount of smoke, sparks flying from the chimney or any deposit in the boiler tubes.

Urquhart's System.*—As the result of mature experience in the use of petroleum refuse as fuel upon the Grazi and Tsaritsin Railway, South-East Russia, Mr. T. Urquhart adopted an arrangement for the firebox which differed slightly from Holden's System described above. He states that "with a locomotive in first-class order, and in the hands of a skilful driver, 50 tons of petroleum refuse are equal to 100 tons of first-class coal." Petroleum refuse was delivered at Tsaritsin, in Russia, at 13*s.* per ton, whilst the price of coal was 27*s.* per ton.†

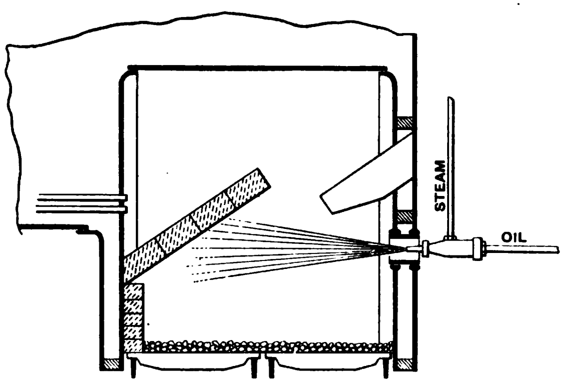


FIG. 29.—Locomotive firebox arranged for oil firing.

Oil Furnace for Lancashire and Water-Tube Boilers.—The Lancashire and Scotch marine types of boilers do not lend themselves to the adoption of oil fuel to the same extent as the water-tube boiler. The necessity for lining the furnace with some refractory material in order to obtain complete combustion of the oil, is in one respect a great disadvantage. In this type of boiler, the front portion of the furnace tube surrounding the furnace forms the most active portion of the heating surface being directly exposed to the radiation from the fire (see Art. 13, Chap. V.). The covering up of this heating surface

* *Proceedings of Institution of Mechanical Engineers*, January, 1889.

† See also a paper on "Petroleum Fuel in Locomotives on the Tehuantepec Railroad of Mexico," *Proc. Inst. Mech. Eng.*, 1906.

with a layer of firebrick makes this portion of the heating surface practically useless for heat transmission, besides making it almost impossible to utilise the direct radiation from the flame. The water-tube boiler furnace is different, being in any case lined with firebrick and of greater volume, and with slight modifications can be rendered suitable for oil firing. The system fitted to a Babcock and Wilcox boiler whereby either

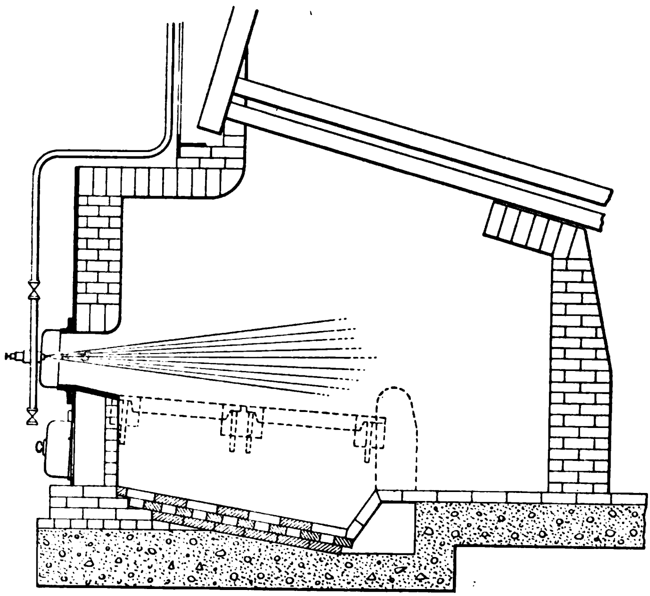


FIG. 30.—Oil fired furnace for Babcock & Wilcox boiler.

liquid fuel or coal may be burned is shown in Fig. 30, the details of the firegrate for burning coal being shown dotted.

Oil fuel has also been used in conjunction with coal to obtain an increased output from existing water-tube boilers. As the result of experiments made at the Westport Station of the Consolidated Gas Electric Light and Power Company of Baltimore,* the arrangement shown in Fig. 31 was adopted.

* A detailed description of this plant will be found in a paper read

As will be seen from the illustration, the space behind the usual coal grate is made into a large combustion chamber with the oil burners at the extreme rear end. This combustion chamber is separated from the boiler tubes above it by tiling, and from the coal grate by a low bridge wall. The bottom of the chamber extends from the burners in a gradual curve to the top of the bridge wall. The hot gases from the burning oil pass over the coal grate and upwards through the usual Babcock and Wilcox setting. Ample air inlets are provided around the burners and

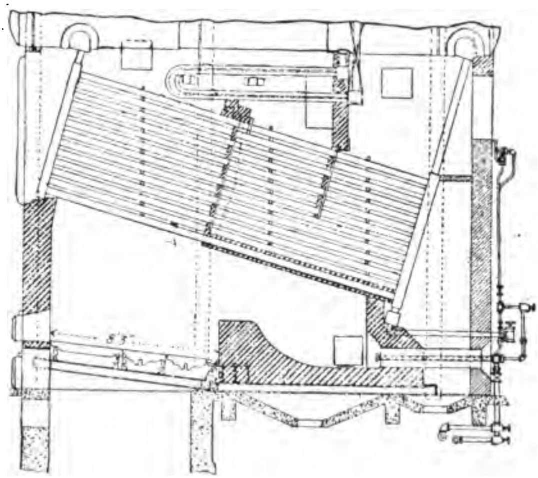


FIG. 31.—Water tube boiler, with setting rebuilt for back-end oil firing.

also in the sides of the combustion chamber. Each furnace is provided with four oil burners, and the coal grates are 14 feet wide and 8 feet long.

The oil is delivered to the burners under a constant pressure of 20 pounds per square inch, and is atomised in each boiler by a steam jet, the steam required for this purpose being under one per cent. of the amount generated by the boiler. By this arrangement either oil or coal or both together may be used to

before the annual convention of the National Electric Light Association, June 1, 1911, by Mr. H. A. Wagner, M.Am.I.E.E.

fire the boiler, and the alteration in the ordinary boiler setting is very slight and requires no additional space. One of the four burners in each furnace is used for the equivalent of a banked coal fire for keeping the boiler ready to steam. These banking burners are operated from a separate oil line, so that a single valve in the main oil line may be used to turn oil into any number of furnaces desired, thus putting any number of banked boilers into use instantly. This arrangement has been found very convenient and effective for meeting the "peak-load" condition which is common to all electric light stations.

18. Gas Furnaces.—Producer gas may be advantageously used in the furnaces of steam boilers employed in works where the gas is manufactured on a large scale, but it would not be advisable, on economic grounds, to make the gas specially for this purpose in spite of the fact that smoke would be entirely prevented. The waste gases from blast furnaces are also sometimes used in boilers, but more frequently in gas engines for driving the necessary blowing engines. Both these gaseous fuels are of low calorific value (about 120 B.Th.U. per cubic foot), since they are largely composed of carbon monoxide, and considerable difficulty is experienced when any attempt is made to burn them in direct contact with the comparatively cool heating surface of a boiler. Combustion is better ensured in the case of a gas of higher calorific value, such as coal gas (of about 600 B.Th.U. per cubic foot), but this would be a most expensive method of steam raising.

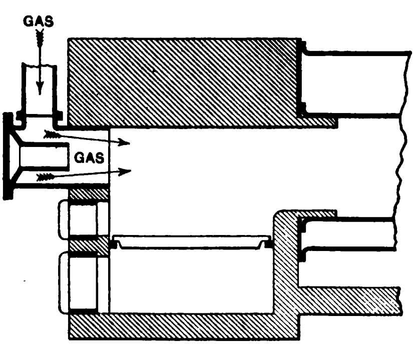


FIG. 32.—Gas furnace for Lancashire boiler.

In order to keep up the high temperature required for efficient combustion a brickwork furnace is used, and so arranged that the combustion is complete before the resulting products

come into contact with the heating surface.* Fig. 32 shows an external brickwork furnace for both gas and coal in connection with a Lancashire boiler; the coal grate may be used either independently or in conjunction with the gas supply, or the gas furnace may be used alone.

A gas furnace fitted to a Babcock and Wilcox boiler at the works of the Normanby Ironworks Co., Ltd., is shown in Fig. 33. The furnace, which is fired with blast furnace gas, is so constructed that the air required for combustion is heated before it meets the gas. The air is admitted under an ordinary coal

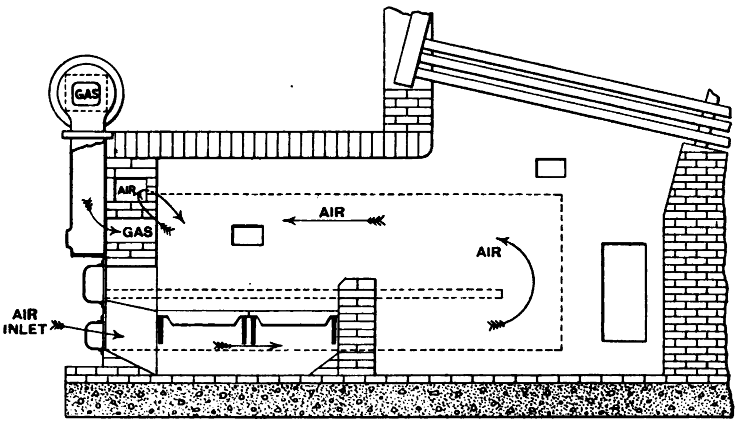


Fig. 33.—Gas furnace for Babcock & Wilcox boiler.

grate, and after passing through brickwork flues surrounded by the hot products of combustion, is delivered into the brickwork combustion chamber through a number of air ducts. The gas also enters the furnace through a number of ports, and by this means the gas and hot air are thoroughly mixed together before combustion takes place. The coal grate is supplied as a stand-by, and also for raising steam when gas is not available, or if required coal may be used in addition to the gaseous fuel.

* For the latest developments in this direction see a paper on Surface Combustion by Prof. W. A. Bone, reproduced in *Engineering*, May 11, 1912.

CHAPTER IV

DRAUGHT

1. Draught.—In order to supply the quantity of air required for the combustion of the fuel in a furnace, it is necessary to maintain a difference in the pressures above and below the fire-grate. This difference of pressure, which is usually measured in inches of water, is known as the *draught*, and may be produced in three ways, namely—

1. By means of a chimney (natural draught).
2. By means of a steam jet (induced or forced draught).
3. By means of a fan, which may either draw the gases from the flues, producing therein a partial vacuum (induced draught), or blow air under pressure into the ash-pit or a closed boiler-room from which the furnace takes its supply (forced draught).

2. Chimney Draught.—Natural or chimney draught is due to the contents of the chimney being lighter than the outside air, *i.e.* to the difference between the weight of the gases inside the chimney and the weight of the same volume of atmospheric air outside the chimney.

When either the whole or a portion of the oxygen in a given weight of air combines with carbon to form carbon dioxide (CO_2), the volume of the products of combustion is the same as the original volume of air supplied at the same temperature; the density, however, is increased in the ratio which the sum of the weights of the air and of the carbon bears to the weight of the air.

When the whole or part of the oxygen in a given weight of air combines with hydrogen to form steam, the volume of the products of combustion is *greater* than the original volume of

air by an amount equal to one-half the volume of the hydrogen burned. Since the quantity of hydrogen in ordinary fuel is such a small proportion of the whole, its effect may be ignored, and it is near enough for all practical calculations to take the volume of the products of combustion at any given temperature as being equal to the volume *at the same temperature* of the air supplied.

The weight of a cubic foot of air at standard atmospheric pressure and at 32° F. being 0·0807 pound, the volume of one pound of air under these conditions is

$$\frac{1}{0{\cdot}0807} = 12{\cdot}39 \text{ cubic feet.}$$

The pressure remaining the same, the volume of a given weight of air is directly proportional to its absolute temperature. Thus at 60° F. the volume of 1 pound of air under standard atmospheric pressure is

$$12{\cdot}39 \times \frac{60 + 461}{32 + 461}$$

$$= 12{\cdot}39 \times \frac{521}{493} = 13 \text{ cubic feet.}$$

Or generally, if V_0 be the volume of a given weight of air at normal temperature and pressure, *i.e.* at standard atmospheric pressure and at 32° F. or $32 + 461 = 493°$ absolute, then the volume of the same weight of air at the same pressure, but at a temperature of $t°$ F., or T° absolute, where $T = t + 461$, will be

$$V_0 \times \frac{t + 461}{493} \quad \text{or} \quad V_0 \times \frac{T}{493}$$

Let w = weight of fuel burned per second in pounds,

n = number of pounds of air supplied per pound of fuel burnt,

$\dot{V}_2$ = volume, in cubic feet, of air supplied per pound of fuel at the ordinary atmospheric temperature T_2 (absolute),

V_0 = volume in cubic feet, at standard atmospheric pressure (14·7 pounds per square inch) and 32° F. of 1 pound of air.

Then from the above it is evident that

$$V_2 = nV_0 \times \frac{T_2}{493} \text{ cubic feet.}$$

Also,

Let T_1 = absolute temperature of the gases in the chimney $= t_1^\circ$ F. $\times$ 461, where t_1° F. is the temperature in the chimney,

$T_0 = 32^\circ$ F. + 461 or 493° absolute,

A = cross-sectional area of the chimney in square feet,

v = velocity, in feet per second, of the gases in the chimney.

Then the total volume of the gases produced per second is

$$w \times nV_0 \times \frac{T_1}{T_0} \text{ cubic feet} \quad . \quad . \quad . \quad . \quad (1)$$

$$v = \frac{\text{total volume}}{\text{sectional area}} = \frac{wnV_0T_1}{AT_0} \text{ feet per second} \quad . \quad (2)$$

and

$$A = \frac{wnV_0T_1}{vT_0} \text{ square feet} \quad . \quad . \quad . \quad . \quad . \quad (3)$$

To allow for air leakage through the brickwork and flues, an increase of some 15 to 20 per cent. should be made in the estimate for V_0 or V_2 over and above the nett amount supplied through the furnace. Under ordinary working conditions t_1 may be reckoned as 600° F. so that $T_1 = 1061^\circ$ absolute.

In using equation (3) to estimate the cross-sectional area of a chimney, the doubtful quantity will be v, the velocity of the gases; it will depend upon the draught.

The draught of a chimney is usually measured in inches of water by a water-gauge, *i.e.* by a glass U tube half filled with water (coloured to render it more easy to read). One vertical leg of the tube is connected by rubber tubing to an iron pipe inserted in the base of the chimney and the end of the other leg is left open to the atmosphere. The pressure on the water in the former leg being lessened by the chimney draught, the pressure of the atmosphere in the other leg drives the water down that leg and up the other. The difference of level of the

two surfaces thus measures the difference of pressure in inches of water.

Since one cubic foot of water at ordinary atmospheric temperature (about 65° F.) weighs 62·3 pounds, its pressure due to each inch of its height is

$$\frac{62 \cdot 3}{12} = 5 \cdot 198 \text{ pound on a base of one square foot,}$$

and of

$$\frac{5 \cdot 198}{144} = 0 \cdot 0361 \text{ pound on a base of one square inch.}$$

The theoretical velocity of the gases is given by the equation,

$$v^2 = 2gl \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

where l is the height of a column of air, measured in feet, corresponding to the draught pressure,

and g is the acceleration due to gravity, viz. 32·2 feet per second per second.

If the draught is h inches of water, the pressure will be 5·198 h pounds per square foot,

and $$l = \frac{5 \cdot 198h}{0 \cdot 0807} \text{ feet.}$$

Hence (4) may be written

$$v^2 = 2 \times 32 \cdot 2 \times \frac{5 \cdot 198h}{0 \cdot 0807}$$

or $$v^2 = 4148h$$

or $$v = \sqrt{4148h} \text{ feet per second} \quad . \quad . \quad . \quad (5)$$

Equation (3) may therefore be used for estimating the cross-sectional area of the chimney, in which w, V_0, T_1 and T_0 will be known, and v is calculated from (5).

It should be remembered that owing to frictional resistance the actual velocity of the gases will be less than that given by equation (5); hence the area calculated from (5) and (3) will be too small. In the case of boilers in which the flues do not exceed 150 feet in length between the firegrate and the base of the chimney, the area of the chimney may be found from the following empirical formula deduced from practice with well-arranged boilers :—

$$A = \frac{G \times W \times C}{\sqrt{H}}$$

where A = smallest internal area of the chimney in square feet,
W = pounds of coal burned per hour,
G = area of firegrate of the boiler or boilers in square feet,
H = height of chimney in feet,
C = a constant varying with the number of boilers discharging their products of combustion into the chimney.

The values of C usually taken are: *

$C = 0{\cdot}100$ for a chimney with 1 boiler.
$C = 0{\cdot}085$ „ „ „ 2 to 6 boilers.
$C = 0{\cdot}075$ „ „ „ 7 to 11 boilers.
$C = 0{\cdot}065$ „ „ „ 12 or more boilers.

The main flue should always be of larger area than the chimney to provide for any reduction in area due to accumulation of soot. The internal surface should be as smooth as possible, and there should be no sudden changes in the cross-sectional area, and no sharp bends in order to reduce frictional resistance. Glazed bricks being not so porous as ordinary bricks minimise any air leakage, but on the other hand are more expensive. A good coating of a tar paint over the brickwork also assists matters. To prevent interference of the draught, the arrangement of the flues from several boilers working on the same chimney, should be such that the various currents of flue gases do not enter the main flue at right angles but are delivered in the same direction as the flow of gases in it. Care is necessary to ensure the dampers shutting closely in the case of one boiler being laid by.

Height of Chimney required to produce a Given Draught.

Let h = required draught in inches of water,
H = height of chimney above the firegrate in feet,
T_1 = absolute temperature inside the chimney,
T_2 = absolute temperature outside the chimney,
n = number of pounds of air supplied per pound of fuel burned.

* From Hutton's "Steam Boiler Construction" (Crosby, Lockwood and Son).

Now, as shown on p. 95, 1 inch of water corresponds to a pressure of 0·0361 pound per square inch, hence a draught of h inches is equal to a pressure of 0·0361 h pounds per square inch, or $0{\cdot}0361 \times 144h = 5{\cdot}198h$ pounds per square foot.

The draught is simply the difference between the weight of a column of external air equivalent in volume to the interior of the chimney and the weight of the same volume of the gases which are actually in the chimney. For convenience, assume the chimney to have a cross-sectional area of 1 square foot, then

volume of gas inside the chimney = H cubic feet.

Neglecting the variation of the density produced by the slightly reduced pressure inside the chimney, we have

$$\left.\begin{array}{l}\text{Density of outside air at}\\ \quad\text{temperature } T_2 \ . \ \ . \ \ .\end{array}\right\} = 0{\cdot}0807 \times \frac{493}{T_2} \text{ pounds per cubic foot.}$$

Here 0·0807 is the weight of 1 cubic foot of air at standard atmospheric pressure (14·7 pounds per square inch) and 32° F. or 493° absolute.

$$\therefore \left.\begin{array}{l}\text{weight of air outside the}\\ \quad\text{chimney equal in}\\ \quad\text{volume to its contents}\end{array}\right\} = H \times \frac{0{\cdot}0807 \times 493}{T_2} \text{ pounds.} \quad (4)$$

Now the density of the gases inside the chimney will be greater than that of the outside air in the proportion $\frac{n+1}{n}$, but much less in the proportion due to the ratio of the temperatures $\frac{T_2}{T_1}$; hence

Density of gases inside chimney

$$= \frac{n+1}{n} \times \frac{T_2}{T_1} \times 0{\cdot}0807 \times \frac{493}{T_2} \text{ pounds per cubic foot.}$$

Weight of gases inside chimney

$$= H \times \frac{n+1}{n} \times \frac{T_2}{T_1} \times 0{\cdot}0807 \times \frac{493}{T_2} \text{ pounds} \quad . \quad (5)$$

Subtracting (5) from (4) we have

$$\mathrm{H} \times \frac{0{\cdot}0807 \times 493}{\mathrm{T}_2} - \mathrm{H} \times \left(\frac{n+1}{n}\right)\frac{\mathrm{T}_2}{\mathrm{T}_1} \times 0{\cdot}0807 \times \frac{493}{\mathrm{T}_2} = 5{\cdot}198h$$

$$\mathrm{H} \times 0{\cdot}0807 \times 493\left(\frac{1}{\mathrm{T}_2} - \frac{n+1}{n} \times \frac{1}{\mathrm{T}_1}\right) = 5{\cdot}198h$$

$$\mathrm{H} \times 0{\cdot}0807 \times 493\left(\frac{n\mathrm{T}_1 - (n+1)\mathrm{T}_2}{n\mathrm{T}_1\mathrm{T}_2}\right) = 5{\cdot}198h$$

$$\mathrm{H} = \frac{5{\cdot}198h}{0{\cdot}0807 \times 493} \times \frac{n\mathrm{T}_1\mathrm{T}_2}{n\mathrm{T}_1 - (n+1)\mathrm{T}_2}$$

$$= 0{\cdot}13h \times \frac{n\mathrm{T}_1\mathrm{T}_2}{n\mathrm{T}_1 - (n+1)\mathrm{T}_2} \quad . \quad . \quad . \quad . \quad . \quad (6)$$

$$= 0{\cdot}13h \times \frac{\mathrm{T}_1\mathrm{T}_2}{\mathrm{T}_1 - \mathrm{T}_2} \text{ approximately} \quad . \quad . \quad . \quad (7)$$

Equation (7) gives the theoretical height of chimney required to produce any given draught in terms of T_1 and T_2.

Equation (7) may also be written

$$h = \frac{\mathrm{H}}{0{\cdot}13} \times \frac{\mathrm{T}_1 - \mathrm{T}_2}{\mathrm{T}_1\mathrm{T}_2} \quad \text{or} \quad \frac{\mathrm{H}}{0{\cdot}13}\left(\frac{1}{\mathrm{T}_2} - \frac{1}{\mathrm{T}_1}\right) \quad . \quad . \quad (8)$$

which gives the draught, in inches of water, produced by a chimney of height H feet when the temperature inside is T_1 absolute, and the temperate outside is T_2 absolute.

In the above theory we have neglected the frictional resistance offered to the passage of the air through the firebars, fire, flues and chimney. In actual practice the effect of this resistance is to reduce the draught h as measured at the base of the chimney below the value obtained from equation (8), or in other words, a greater height of chimney will be required to produce a draught h than is obtained from equation (7). The actual draught by water-gauge in a mill chimney varies from $\frac{1}{2}$ to $\frac{3}{4}$ inch of water at the base of the chimney.

Again, this air frictional resistance is not a constant quantity for all chimneys; it depends upon the velocity with which the gases pass through the flues and chimney, and therefore, also upon the thickness and density of the fire and the width of the air-spaces between the firebars. In addition to these factors

the resistance is proportional to the height of the chimney and to the "hydraulic mean depth," *i.e.* the ratio

$$\frac{\text{cross sectional area of chimney}}{\text{perimeter of cross section}}$$

If therefore we increase the height of the chimney in order to allow for the frictional resistance, it is obvious that the frictional resistance will increase also. It does not therefore follow that increasing the height of a chimney will always increase its draught; if the velocity of the gases through it is too high so that the loss of head due to the frictional resistance of the chimney is high as compared with the loss due to the frictional resistance through the firebars and flues, it may happen that the additional head due to an increase in height may not counterbalance the increased frictional resistance. It must be remembered that through radiation, the temperature of the gases in the chimney is in reality diminishing for every foot in its height, and their actual mean temperature is therefore less than T_1 at the base. Additional height is therefore only of use if the chimney is already amply large enough in area, so that the additional suction is really spent over the firegrate and flues.

Seeing that the resistance depends upon the "hydraulic mean depth" whilst equation (7) is independent of the dimensions of the cross section, it will be readily understood that the attempt to use a formula obtained from purely theoretical considerations is beset with difficulties.

Many empirical formulæ have been given from time to time of which that due to Kent * is perhaps the simplest, namely:—

$$H = \left(\frac{0{\cdot}06\ W}{A}\right)^2$$

where H = height of chimney in feet,

W = pounds of coal burned per hour,

A = effective cross sectional area of the chimney in square feet,

$= \text{smallest area} - 0{\cdot}6\sqrt{\text{smallest area}}.$

The height of a boiler chimney must at least be sufficient to carry away the gases without causing any nuisance to the

* *Proc. Am. Soc. M.E.*, vol. vi.

neighbourhood, and it may be governed by the by-laws of the district. It is seldom less than 70 feet, but factory chimneys vary in height from about 150 feet to 200 feet.

3. Mechanical Draught.—To obtain the best results from the use of a mechanically produced draught it is desirable that the necessary accessories should be fitted to the boiler when first installed, and not, as is frequently the case, applied to a boiler which has been designed and constructed for working with natural chimney draught. The function of a steam boiler is to generate steam at the cheapest rate possible; this, of course, means that the running cost, *i.e.* cost of fuel and wages, and also the charges for maintenance, upkeep, and repairs should be reduced to a minimum. If a mechanical draught is fitted to a boiler whose furnace is unsuitable, there is a much greater danger of increased expenditure in upkeep and repairs, than if the boiler and furnace were in the first place designed to work with such a draught.

With a mechanical, or forced, draught, it is possible to supply a larger quantity of air to the furnace and therefore to burn more coal per square foot of grate area than is possible with chimney draught. This is due to the fact that a greater difference in the pressures above and below the firegrate can be maintained by mechanical means than by a chimney. To obtain proper combustion it is necessary that a high furnace temperature should be maintained, and in order to secure this the furnace should be of large capacity and lined with firebrick, and so arranged that combustion is complete before any of the gases come in contact with the relatively cold heating surface of the boiler (Art. 1, Chap. III.). A high furnace temperature is the natural outcome of the use of forced draught, and, in addition, it is possible to regulate the supply of air and obtain complete combustion with less air per pound of coal than is possible with natural chimney draught. This, of course, results in a smaller weight of flue gases per pound of fuel burned, and therefore, on this account alone, a reduction in the heat lost in the flue gases.

Owing to the high furnace temperature obtained with forced draught there is a higher rate of heat transmission through the boiler plates to the water, and consequently the heating surface

of the boiler *may* be reduced. If this is done, the temperature of the gases leaving the boiler may be as high as 900° F., and it will be found cheaper to install economisers to further cool the gases down to a temperature not much above that of the boiler steam rather than to extend the heating surface of the boiler. This will result in a further saving in the quantity of heat carried away by the flue gases.

From the above considerations it is evident that the most economical results can be obtained from a steam boiler, when a strong, mechanically produced draught is used, providing that the heating surface is extended by means of an economiser in order to cool the gases to as low a temperature as practicable before they escape, and that the minimum amount of air is supplied which will give complete combustion.

In many cases where forced draught is used in practice no special attempt is made to cool the gases down to a low temperature before escaping. Notable examples exist in the case of locomotive and marine boilers, where the chief reason for using forced draught is to burn more fuel per square foot of grate area, obtaining thereby a higher furnace temperature and more steam generated in a boiler of given weight. The result in such cases is, of course, a lower efficiency due to the loss occasioned by allowing the gases to escape at a high temperature.

Induced draught has the advantage of being in most cases easily applied to existing boilers at a much cheaper rate than forced draught, but special care should be taken to prevent air-leakage into the boiler flues, and a very good plan to prevent this is to cover the whole of the brickwork with light sheet-iron plates. By this means greater economy can be obtained. Since, however, the fan works in the hot gases, it takes more horse-power than a fan dealing with cold air as in forced draught.

If h be as before the water-gauge reading in inches, the pressure or suction is $5{\cdot}2\,h$ pounds per square foot; if V be the volume of air or gases passing through the fan in cubic feet per minute, the net horsepower expended in driving the air is

$$\frac{5{\cdot}2\,h \times V}{33{,}000}$$

and if η = the efficiency of the fan, the brake horsepower of the driving engine or motor is

$$\text{B.H.P.} = \frac{5 \cdot 2h \times \text{V}}{33{,}000 \times \eta}$$

Let n pounds of air be supplied per pound of fuel burned, and let w be the weight of fuel burned per second in pounds.

With forced draught suppose that this is supplied at a temperature of 62° F. Then

$$\text{V}_f = 60\, nw\, \text{V}_0 \times \frac{62 + 461}{493}$$

where V_0 is the volume of one pound of air at normal temperature and pressure.

After passing through the furnace there will (neglecting the proportion of ash) be nearly $(n + 1)w$ pounds of flue gases delivered per second, consisting, say, of 80 per cent. nitrogen, 12 per cent. carbon dioxide and 8 per cent. oxygen. Since the cubic feet per lb. of nitrogen, carbon dioxide and oxygen at 32° F. are respectively 12·75, 8·15 and 11·2, there is no great error in again assuming for the mixture the same volume per pound at 32° F. as for air, *i.e.* $\text{V}_0 = 12 \cdot 39$ cubic feet. With induced draught the temperature at which the gases enter the fan may be taken as 350° F. Hence

$$\text{V}_i = 60(n + 1)w\, \text{V}_0 \times \frac{350 + 461}{490}$$

The horsepower of the induced draught fan as compared with the horsepower of the forced draught fan is therefore increased in the ratio

$$\frac{\text{V}_i}{\text{V}_f} = \frac{n + 1}{n} \times \frac{811}{523}$$

or if $n = 18$ pounds $\frac{\text{V}_i}{\text{V}_f} = 1 \cdot 64$, an increase of 64 per cent. Even this does not fully express the difference, since in the case of the induced draught $(n + 1)w$ must be increased by some 20 per cent. to cover the air which leaks into the flues, etc., through the brickwork, so that V_i is not far from double V_f.

A larger and special type of fan is therefore required, the working costs and upkeep being generally greater than with

forced draught. With forced draught, in which a fan is used to force air into the furnace, there is the expense of the air ducts from the fan to the furnace, but the pressure being above atmospheric there is no loss from air leakage into the flues and no rush of cold air into the furnace when firing by hand, such as exists with induced draught. It is, however, usual when boilers are hand-fired to fit dampers so that when the furnace doors are opened this rush of cold air into the furnace does not occur, nor in the case of forced draught, a rush of flame outwards. Which of the two systems, induced or forced draught, is the best for large installations it is difficult to say, but the probability is that forced draught is the most successful when put down in conjunction with new boilers.

A comparison of either system with chimney draught as regards their respective waste of heat is also of interest. The loss of heat per lb. of coal if t_1 is the temperature of the gases as they enter the chimney or fan is

$$1 \text{ to } 1{\cdot}2(n+1) \times (t_1 - t) \times 0{\cdot}238 \text{ B.Th.U.}$$

where 0·238 is the mean specific heat of the gases. Assuming for the sake of comparison the same consumption of coal per minute, but a supply of 24 lbs. of air with chimney draught as against 18 lbs. with mechanical draught, and a temperature of 600° F. at the base of the chimney as against 350° F. for the gases leaving the flues, the loss of heat per lb. of coal is

$$\begin{aligned}
\text{with chimney draught} &= 1{\cdot}15 \times 25 \times (600 - 62) \times 0{\cdot}238 \\
&= 3700 \text{ B.Th.U.} \\
\text{with induced draught} &= 1{\cdot}2 \times 19 \times (350 - 62) \times 0{\cdot}238 \\
&= 1560 \text{ B.Th.U.} \\
\text{with forced draught} &= 19 \times (350 - 62) \times 0{\cdot}238 \\
&= 1300 \text{ B.Th.U.}
\end{aligned}$$

out of a total of say, 14,000 B.Th.U. Against these savings of 2140 and 2400 B.Th.U. (about 15 and 17 per cent.) per lb. of coal must of course be debited the cost of the power employed in driving the fan.

The increased weight of steam obtained by using forced draught on board ship enables fewer boilers to be used and therefore less dead weight has to be carried. Also since the

draught produced is independent of the temperature of the atmosphere, it is maintained at the same figure whether in the tropics or in cooler regions, which would be impossible if chimney draught was used (see preceding Art. 2). In addition to this, better ventilation is obtained when induced draught is used than would be possible with any other system.

When the demand for power increases to such an extent that it is found necessary to install additional boilers in an existing steam plant, it may happen that the chimney is too small to create the necessary draught. In such cases a simple solution to the difficulty is found by fitting a fan in the base of the chimney.

The advantages of using a mechanically produced draught may be briefly summed up as follows: Increased evaporation per square foot of heating surface; higher furnace temperatures and consequently increased efficiency; small chimneys may be used; cheaper coal can be economically burned; the easier regulation of the output of steam and the ability to meet sudden demands such as exist in electricity supply stations, rolling mills, warships, etc.; no dependence need be placed on the weather. The chief disadvantage of the system lies in the heavy upkeep and repairs that must be expected owing to the increased duty which the boilers are called upon to perform.

4. Steam Jets.—Instead of using a fan in the base of the chimney to produce the partial vacuum required, steam jets may be employed, but with a lower economy on account of their much greater steam consumption.

In addition to the lower efficiency resulting from their use, steam jets have the further disadvantage of increasing the tendency to external corrosion, particularly when using fuel containing an appreciable amount of sulphur. The combustion of sulphur results in the formation of sulphur dioxide (SO_2), which in the presence of moisture forms sulphurous acid which is an active corrosive agent (see also p. 27).

Steam jets are much cheaper in first cost than fans, and in land practice are frequently used in a closed ashpit for creating a forced draught of moderate intensity. In many cases steam

jets are employed under the firebars in an open ashpit, not to produce a draught, but to keep the bars cool and so reduce the labour of removing clinker when cleaning the fires. In some cases, when burning cheap bituminous coal on a mechanical stoker, their use results in an appreciable reduction in the amount of smoke emitted from the chimney.

With the single exception of the locomotive, the use of steam jets in the ashpit always results in lower economy because of the increased specific heat of the flue gases which results in a greater loss up the chimney. In other words, as explained on p. 34, the loss of heat occasioned thereby will be that due to raising the steam from its own initial temperature to the temperature of the flue gases. It will be obvious that this loss does not occur in the case of locomotives, because the exhaust steam from the engine cylinders blows directly up the funnel without passing through the fire at all; further the quantity of exhaust is proportional to the load on the engine, and the greater the load, the greater the quantity of steam and the greater the draught; in this way, therefore, the draught is more or less self regulating.

5. Closed Stokehold.—In this system, used almost exclusively in the Navy, the necessary difference of pressure above and below the firegrate is maintained by the use of fans which deliver a uniform supply of air under pressure into the stokehold. By this means the pressure inside the combustion chamber is always below the air pressure in the stokehold—as is the case with chimney draught—and there is no outrush of the flames into the stokehold when the furnace doors are opened for firing. It will be evident that the stoke-hold must be airtight, and therefore shut off from the rest of the ship, in order that the air pressure in it may be maintained constant when the fans are working.

Under easy load it is not necessary, as a rule, to work the fans since the funnel itself may be able to maintain the required draught; under these conditions, therefore, the stokehold is open and not isolated from the rest of the ship. It is only in cases of emergency when every boiler is required to work at its maximum power that the stokehold is closed and the fans used.

6. Closed Ashpit.—In this system the ashpit only instead of the whole stokehold is closed; fans deliver air under pressure into the closed ashpit. Special inter-connected furnace doors and dampers should be fitted so that the act of opening the former closes the dampers and so prevents the outward rush of flames which would otherwise scorch the stoker. Closing the doors opens the dampers and renews the air supply.

In *Howden's system* of forced draught, the air supplied to the furnace is heated by passing round a series of tubes through which the hot furnace gases circulate. The ashpit is closed and the hot air is supplied both above and below the firegrate.

Assuming a temperature for the funnel gases in the uptake of, say, 400° F., and 16 pounds of air per lb. of coal, the saving from heating the air from 70° to 350° will be roughly $16 \times (350 - 70) \times 0{\cdot}24 = 1070$ B.Th.U. per lb. of coal burnt. But this is not the end of the gain, since the temperature reached by the furnace gases will also be increased and may rise say from 2400° F. with cold air to 2700° F. with the heated supply.

Figs. 34 and 34A show the application of Howden's patent hot-air forced-draught system to a marine boiler on board ship.

Referring to these figures it will be noticed that the air is delivered under pressure from the fan and, passing along the air trunk shown, is led to the front end of the boiler, where it circulates round a vertical nest of tubes placed in the uptake. The hot furnace gases pass through these tubes on their way from the smoke box to the funnel, and in doing so raise the temperature of the surrounding air. The hot air is then led down the front of the boiler and delivered both above and below the firegrate: the path taken by the air is clearly shown by the arrows.

7. Ellis and Eaves' System.—This consists of an induced draught system in conjunction with hot-air economisers which utilize the waste heat in the gases as they leave the boilers for heating up the air required for combustion; a fan is placed at the base of the chimney and draws the gases as they come from the boiler through a series of tubes which form the hot-air economiser. The fan draws in the air required for combustion through apertures in the air-heater; this air circulates

round the tubes through which the hot gases are passing, and is raised to a temperature of about 300° F. before entering the furnace. Messrs. Ellis and Eaves have recently introduced an improved system called the "Atlas" system of balanced draught, this method being a combination of forced and induced draught with air-heating economisers. Two fans are employed, one suction and the other pressure, preferably mounted on one shaft and driven by a single engine or motor. The suction-fan draws the waste products of combustion through the tubes of the air-heating economiser, while the pressure-fan forces the air for combustion around the outside of these tubes to heat it before passing to the furnace. The fans are so proportioned that the vacuum set up by the suction-fan is just sufficient to overcome the resistance due to the flues, tubes, etc., when the pressure-fan is maintaining the requisite pressure under the firegrate to burn the required amount of coal. The draught in the furnace is thus balanced so that there is no rush of flame outwards or inrush of cold air when the furnace doors are opened.

8. **Dampers.**—In order to regulate the draught to suit the load, dampers are always fitted in the main flues of every boiler; they also provide an efficient means of isolating any boiler when it is shut down because, when fully closed, communication between the furnace and chimney is completely prevented. Such dampers will, in some cases, effectually control the rate of combustion to suit the load on the boiler. For instance, if the steam pressure rises above a certain fixed value, a partial closing of the main dampers will reduce the rate of combustion until the pressure falls; similarly, if the pressure falls below a certain fixed value due to an increased demand for steam, a further opening of the damper will enable the pressure to rise and be maintained at its proper value, provided of course the boiler is not overloaded and the fall of pressure not produced by the fires becoming clinkered up. In some cases the rise or fall of the steam pressure is, by a suitable mechanism, made to close or open the dampers automatically.

A damper in the main flue leaving a boiler is, however, not always sufficient to control the air supply efficiently. In the

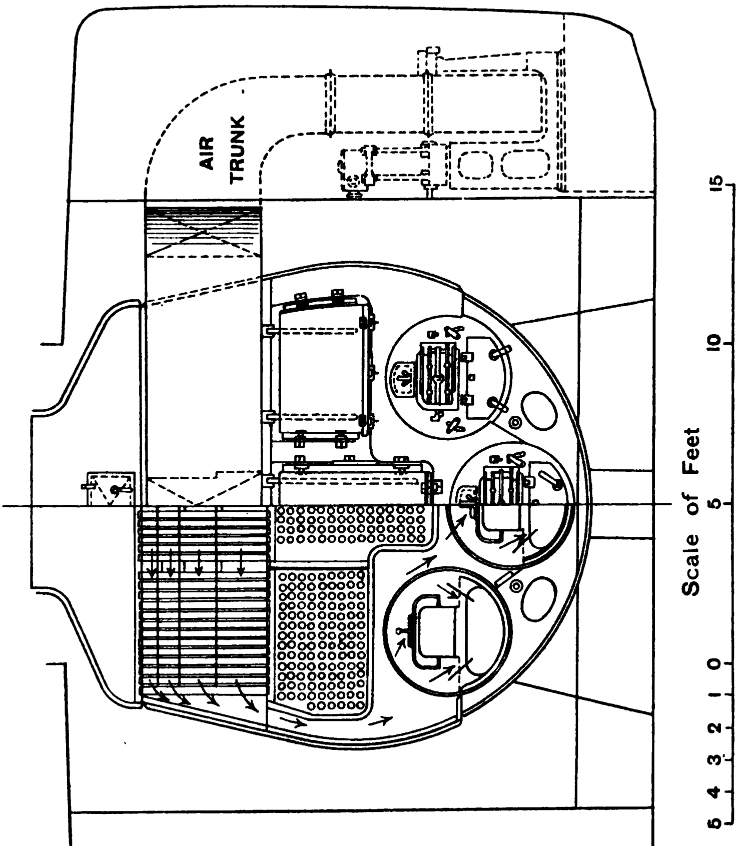


Fig. 84.—Howden's system of forced draught (transverse section).

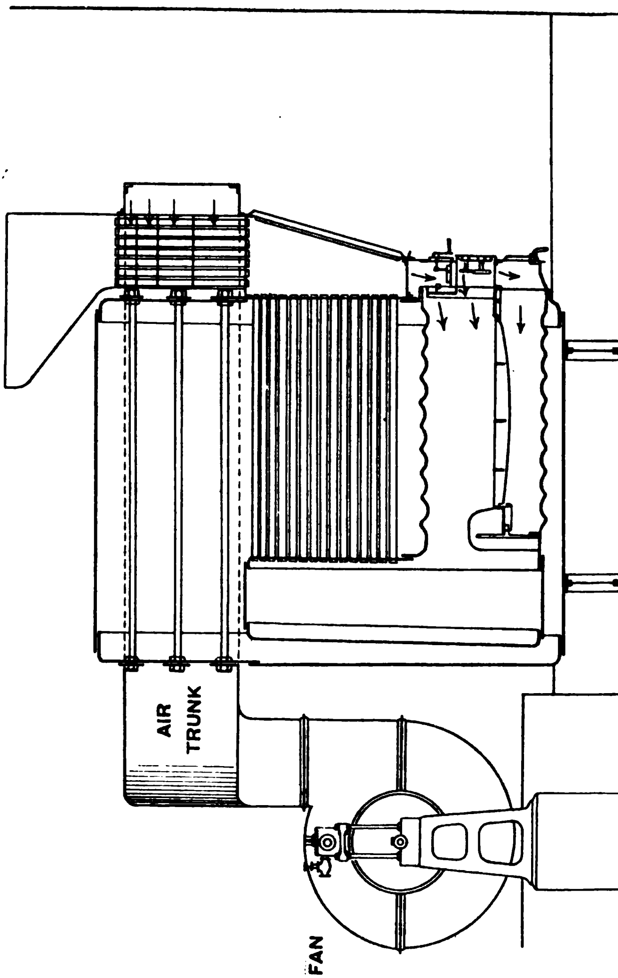


Fig. 34a.—Howden's system of forced draught (longitudinal section).

case of a Scotch marine boiler with separate combustion chambers, the closing of the main damper will obviously reduce the air supply to all the furnaces of that boiler. This is not always desirable because the condition of the separate fires may be entirely different, some requiring more air than others. In such cases a damper is often fitted at the exit from each combustion chamber in order that the draught may be regulated in each furnace, particularly when forced draught is used in conjunction with closed ashpits.

In addition to the damper in the main exit flue from a boiler, dampers are often fitted near the entrance to the ashpits; this still leaves it possible to regulate the air supply through the grids in the furnace doors quite independently of that through the air-spaces between the firebars. This is an important matter when burning bituminous coal because, after firing, the combustion of the hydrocarbons given off depends upon the air admitted above the fire as already explained in Arts. 4 and 6, Chap. III. If only the damper in the main exit flue is used, any closing of it will obviously reduce the air supply both above and below the firegrate, whereas if the damper at the entrance to the ashpit be used, either alone or in conjunction with the main damper, the air supply above the fire may still be maintained of sufficient amount to ensure a better chance of smokeless combustion.

The type of damper used depends upon the size of flue in which it is to work. In small or narrow flues, *e.g.* the side flues of a Lancashire boiler, dampers are used which slide in vertical grooves in the walls of the flue, whereas in large wide flues swivel dampers are used, which rotate about pivots attached to their centre line.

9. Effect of Draught on the Limits of Combustion.—With ordinary coal the maximum rates of combustion (in pounds per hour per square foot of grate area) possible with various types of boilers are approximately

With chimney draught and any type of boiler about 24 pounds.
With forced draught in Lancashire and Scotch marine boilers . . . . „ 30 „

With forced draught in Yarrow and Thornycroft boilers as fitted in torpedo boats . . about 60 pounds
With locomotives . . . . „ 120 „

The exceptionally high rate with a locomotive running on the road (120 pounds) is only obtained at the expense of low efficiency; unburnt cinders are drawn through the tubes and deposited in the smoke box or thrown out of the funnel by the high draught.* As much as 70 pounds per hour per square foot of grate area have also been burned in the Babcock and Wilcox boilers of U.S. battleship Utah, the resulting efficiency, however, being reduced to about 60 per cent.†

It should be remembered that with all boilers, the extent to which the rate of combustion may be forced depends solely upon the amount of heat that the heating surface is capable of transmitting to the water. The stronger the draught, the higher is the velocity with which the hot gases move over the heating surface and consequently the greater the rate of heat transmission (Art. 16, Chap. V.). From experiments carried out by one of the United States Government Departments it was found that although the rate of heat transmission was increased due to the velocity of the gases, the circulation in water-tube boilers did not increase in the same proportion owing probably to the increased friction set up between the water and the tubes; in this may be found the reason why this type of boiler gets overheated and tubes are burnt out when doing high duty. In fact in the British Navy, owing to this cause, Babcock boilers are only worked up to burn about 35 pounds of coal per hour per square foot of grate area. Much higher rates of combustion have been obtained with boilers of the Yarrow and Thornycroft type (see above), but it is noticeable that these small-tube boilers have a very rapid water circulation.

Another important point to be remembered in connection with high-duty mechanical-draught plant, is the quality and

* In the case of an express engine the draught may be as high as 10 inches of water. See *Proc. Inst. Mec. Eng.*, 1908, p. 262.

† See paper by Mr. E. M. Speakman on "The Wider Adoption and Standardisation of Water-tube Boilers," *Inst. Engineers and Shipbuilders in Scotland*, Feb., 1912.—*Engineering*, March 8, 1912.

size of the coal to be used since this has a great bearing on the efficiency and evaporative power of the boiler. When mechanical stokers are adopted, the necessity of using self-cleaning stokers for mechanical draught is demonstrated by the fact that in ordinary practice it is necessary to clean the fires after using about 2 cwt. of coal per square foot of grate, so that when burning about 50 or 60 pounds of coal per hour per square foot of grate area, the fires would require cleaning every 3½ to 4½ hours, which would mean having to use stand-by boilers during the operation.

10. The type of Boiler best suited for Mechanical Draught.—In this connection it would appear that all boilers of rigid construction, such as the Lancashire and Scotch marine types, suffer very badly from the high temperature and the heavy stresses thereby set up, and from this cause appear to be undesirable for strong draughts at any rate. Water-tube boilers, however, do not suffer to the same extent, and are thus more suitable if a high duty is to be maintained. The furnace of a water-tube boiler meets the requirements laid down in Art. 7, Chap. III., being of large capacity and always lined with firebrick, so that it is eminently suitable for the high-furnace temperature obtained. It appears, therefore, that when installing any mechanical draught steam plant the following points should be considered if good results are to be obtained:—If suitable for the working conditions required (see Art. 3, Chap. I.), a water-tube boiler with a rapid water circulation should be used, and a large and properly designed firebrick-lined furnace should be fitted. Self-cleaning mechanical stokers should be used to avoid the repeated cleaning of the grates and the opening of furnace doors. To produce the necessary draught an efficient and high-class fan should be used. All boiler feed-water should be treated (Chap. X.) before being pumped into the boiler and a suitable feed-water heater or economiser employed to raise the water to as nearly as possible the same temperature as the steam before it enters the boiler.

CHAPTER V

GENERATION OF STEAM AND HEAT TRANSMISSION

1. Effect of Heat on Water.—In order to obtain the best results, heat should be applied to the bottom of the vessel containing water; the particles of water at the bottom are then heated first and becoming less dense they ascend, colder particles flowing down to take their place. This action continues until all the water is at the same temperature, the result being a continuous circulation of the water, one current travelling upwards and another downwards. By means of these *convection* currents, as they are called, heat is rapidly transmitted throughout the mass of water. In the case of a steam boiler the water circulation is of great importance; the quicker the circulation, the more rapidly is heat transmitted throughout the contents of the boiler and therefore the more rapidly can steam be generated.

When the temperature is high enough, steam will be formed on the water side of the heating surface, and the upward convection current will consist of a mixture of steam bubbles and hot water, the downward current consisting of cooler water only. The water space in a steam boiler should be designed to give ample area for both the ascending and descending currents, which should be kept separated in order to facilitate the water circulation.

The circulation of water in a steam boiler, then, is due to the ascending and descending convection currents which are

produced and maintained by two agencies, namely, the variations in the temperature of different portions of the water, and the presence of steam bubbles which reduce the density of the mixture.* In the ideal boiler, the steam bubbles are swept away from the heating surface as they are formed and the greatest amount of steam is generated from a given weight of fuel. Given an existing heating surface maintained at a sufficiently high temperature, the quantity of heat which can be transmitted to the water is only limited by the rate at which it can be carried away from the heating surface by the convection currents.

The steam when formed has a natural tendency to cling to the heating surface of a boiler, and when the water spaces do not permit the ascending and descending convection currents to be kept separate, the circulation is impeded and the steam accumulates on the heating surface forming a film between the plate and the water. This may result in the plates getting overheated, and in the gauge glasses showing a higher water level than that due to the natural position of the steam and water.

When the plates become overheated, the water in the boiler may not wet the plate but may roll on its surface in drops, being separated from the plate by a layer of steam. This tendency is facilitated by the presence of grease in the feed water; hence the importance of removing the oil, etc., from the condensed exhaust steam from the engine, when it is used as hot feed water to the boiler.

The steam bubbles are liberated at the surface of the water and pass into the steam space above the water. When the height of the water in the boiler is at its normal working level, the area of the water surface should be large enough to allow the bubbles to escape quietly so as to obtain a steady supply of dry steam without inducing priming. If the area of the water surface is too small the steam is released more violently, and wet steam and priming may be the result.

The higher the pressure at which the steam is generated, the less may be the water surface because the volume of the

* For the results of experiments on the Circulation in Water-tube Boilers, see a paper by Prof. W. H. Watkinson, *Proc. Inst. N.A.*, 1896, vol. xxxvii., p. 267.

same weight of steam is reduced. If it is compulsory to use dirty feed water the water surface should be very much larger than when clean soft feed water is used.

2. Saturated Steam.—Steam formed in contact with water is known as *saturated* steam and may be either dry or wet. *Dry* saturated steam consists only of water vapour, whilst *wet* steam is dry steam containing particles of moisture mechanically suspended in it. An analogy will make this clearer. The atmosphere on a clear day may be compared with dry saturated steam, and on a misty day with wet steam; the atmospheric air may be considered as dry steam, while the mist is produced by small particles of water mechanically mixed with and floating about in the air, and the mixture may be likened to wet steam.

3. Superheated Steam.—Superheated steam approximates more closely to a perfect gas than does saturated steam, and for this reason has been called "*steam gas.*" It is produced by heating dry saturated steam until its temperature is higher than the temperature at which the saturated steam was produced. Saturated steam at a given pressure can have but one temperature, known as the temperature of saturation; by further applying heat to it there is no limit (other than that imposed by practical considerations) to the temperature to which the superheated steam may be raised, even if the pressure remains constant during the operation.

Temperature and Pressure of Saturated Steam.—The temperature at which steam is formed from water depends upon the value of the pressure to which it is subjected. For instance, at standard atmospheric pressure (14·7 pounds per square inch absolute) the temperature of saturation is 212° F.; at a pressure of 2 pounds per square inch absolute, the temperature is only 126° F.—whereas at a pressure of 150 pounds per square inch absolute, the temperature is 358° F. Equal increments in the pressure, however, do not give rise to equal increments in the temperature. By referring to the steam tables on p. 407 it will be seen that equal increments of 50 pounds in the pressure from

20 pounds per square inch absolute the temperatures are as follows:—

Pressure lbs. per sq. in. absolute.	Temperature ° F.	Increase in temperature ° F.
20	228	
		75
70	303	
		38
120	341	
		27
170	368	

As the pressure increases the temperature increases also, but at a slower rate, or in other words, the pressure of saturated steam rises with the temperature at a rate which increases rapidly the higher the temperature rises. There is no simple law connecting pressure and temperature, the results given above having been obtained experimentally. Rankine expressed the relation by the empirical formula

$$\log p = 6{\cdot}1007 - \frac{2732}{T} - \frac{396945}{T^2}$$

where p is the pressure in pounds per square inch, and T the absolute temperature in degrees Fahrenheit.

4. Relation of Pressure and Volume in Saturated Steam.—The volume V occupied by one pound of dry saturated steam at different pressures is called the *specific volume* and is shown in the steam tables, being calculated from the formula

$$V = w + \frac{JL}{T} \cdot \frac{dT}{dP}$$

where w = volume in cubic feet of 1 pound of water (0·016 cubic foot),

L = latent heat of steam in British Thermal Units per pound at absolute temperature T Fahrenheit,

J = mechanical equivalent of heat = 778,

$\frac{dT}{dP}$ = rate of increase of T with pressure P reckoned in pounds per square foot.

5. Total Heat of Steam.—Suppose we have one pound of water at 32° F. and apply heat to it at constant standard atmospheric pressure, 14·7 pounds per square inch. The temperature will rise until it reaches the temperature of saturation (or the boiling point), namely, 212° F. The amount of heat supplied will be very nearly $212 - 32 = 180$ British Thermal Units. The quantity of heat so supplied, which raises the temperature of the water, is called *sensible heat*. If now we continue supplying heat to the pound of water at 212° F., the water will be turned into steam *at the same temperature*, and until all the water is evaporated its temperature will remain at 212° F. When all the water is evaporated, it will be found that an additional 966·6 British Thermal Units * have been supplied. The quantity of heat so supplied, which changes the physical state of the water from liquid to vapour without changing its temperature, is called the *latent heat* of evaporation.

The *total heat of evaporation* of the steam in the above case, is the total amount of heat supplied to the pound of water at 32° F., in order to turn it all into dry saturated steam at 212° F., being equal to the sum of the sensible and latent heats, or

$$\begin{aligned}\text{Total heat (H)} &= \text{sensible heat } (h) + \text{latent heat (L)}\\ &= 180 + 966{\cdot}6\\ &= 1146{\cdot}6 \text{ B.Th.U.}\end{aligned}$$

The total heat of steam is always reckoned from water at 32° F., and in general, when dry saturated steam is produced at a temperature t° F., we have

$$\begin{aligned}\text{Total heat (H)} &= \text{sensible heat } (h) + \text{latent heat (L)}\\ \text{H} &= h + \text{L} \qquad \qquad (1)\\ &= (t - 32) + \text{L}\end{aligned}$$

* Recent research has shown that the latent heat of steam at 212° F. is more nearly 970 than 966·6 B.Th.U. The latter figure is, however, here provisionally retained, since it is still largely used and forms the basis of many well-known empirical equations for the heat of steam. The higher figure is adopted in the Steam Table (p. 406) with the consequent modification that it entails, so that the Table without being strictly accurate to a degree of refinement unnecessary in boiler work may be taken as representing approximately the results of later experiments.

6. Latent Heat.—When steam is generated at constant pressure, the total heat of evaporation as defined above is *not* actually present in the steam. While evaporation is taking place, work has to be done against the pressure in order to allow the change in volume to take place as the water changes into steam. An example will make this clear. Consider a pound of water under the constant atmospheric pressure of 14·7 pounds per square inch and at a temperature of 212° F.; it will occupy a volume of 0·016 cubic foot. When it has all been evaporated, the pound of dry saturated steam which results will occupy a volume of 26·6 cubic feet. The *external* work done in foot pounds against the pressure of 14·7 pounds per square inch, or $14{\cdot}7 \times 144 = 2116$ pounds per square foot will therefore be

Pressure (pounds per square foot) × change in volume (cubic feet)

$$\begin{aligned} &= 2116 \times (26{\cdot}6 - 0{\cdot}016) \\ &= 56{,}252 \text{ foot pounds} \\ &= \frac{56{,}252}{778} = 72{\cdot}3 \text{ B.Th.U.} \end{aligned}$$

since 778 foot pounds are equal to 1 B. Th. U.

Now the quantity of heat which has been supplied to evaporate the water, *i.e.* the latent heat, is 966·6 B.Th.U. (Art. 5), and the difference $966{\cdot}6 - 72{\cdot}3 = 894{\cdot}3$ B.Th.U. is known as the internal latent heat usually denoted by the Greek letter ρ.

Let P = pressure (absolute) in pounds per square foot at which the steam is generated,

V = volume (in cubic feet) of 1 pound of dry saturated steam at pressure P,

w = volume (in cubic feet) of 1 pound of water = 0·016 cubic foot.

Then,

External work done during evaporation (E) $= P(V - w)$ foot pounds

$$= \frac{P}{J}(V - w) \text{ B.Th.U.}$$

where J is the mechanical equivalent of heat = 778 used above, and

Internal latent heat = latent heat − external work done

$$\rho = L - \frac{P}{J}(V - w) \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

The *internal or intrinsic energy* of the steam is the name given to the sum of the internal latent heat and the sensible heat of evaporation, and it is this which is actually present in the steam, *i.e.*

$$\text{internal or intrinsic energy} = \rho + h$$
$$\text{and total heat of evaporation, } H = h + L$$
$$= h + \rho + E \quad . \quad . \quad . \quad (2)$$

The latent heat of steam varies with the temperature at which it is produced and may be calculated from the empirical formula,

$$L = 1114 - 0{\cdot}7t \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

where t is the temperature of saturation in degrees F.

The total heat of dry saturated steam at temperature t° F. may therefore be approximately expressed as

$$\begin{aligned} H &= h + L \\ &= (t - 32) + 1114 - 0{\cdot}7t \\ &= 1082 + 0{\cdot}3t \text{ B.Th.U. per pound} \quad . \quad . \quad . \quad (4) \end{aligned}$$

It will be observed from equation (3) that the latent heat *decreases* as the temperature (and therefore the pressure) increases, but from (4) the total heat always *increases* as the temperature (and therefore the pressure) increases.

7. Total Heat of Superheated Steam.—If heat be applied at constant pressure to one pound of dry saturated steam at temperature t° F., superheated steam is the result, and the temperature will rise to, say, t_1° F. The sensible heat so supplied will be,

Specific heat of steam at constant pressure × rise in temperature

$$= 0{\cdot}48 \times (t_1 - t) \text{ B.Th.U.} \quad . \quad . \quad . \quad . \quad . \quad (5)$$

The specific heat of steam at constant pressure is not a constant quantity; at atmospheric pressure and in the neighbourhood of 212° F. it is about 0·48, but it increases with the pressure until at 150 pounds per square inch it is about 0·6.

The total heat of superheated steam may therefore be expressed by the approximate relation,

$$H = 1082 + 0{\cdot}3t + 0{\cdot}48(t_1 - t) \text{ B.Th.U.} \quad . \quad . \quad (6)$$

where t_1 is the temperature of the superheated steam, and t the temperature of saturation corresponding to the pressure of the steam, both being measured in degrees Fahrenheit.

Example.—Find the total heat of one pound of superheated steam at 200 pounds per square inch absolute and 500° F.

From the steam tables (p. 407) we see that the temperature of saturation corresponding to a pressure of 200 pounds per square inch is 381·5° F.

Hence

$$\begin{aligned} H &= 1082 + 0{\cdot}3 \times 381{\cdot}5 + 0{\cdot}48(500 - 381{\cdot}5) \\ &= 1082 + 114{\cdot}45 + 56{\cdot}88 \\ &= 1253{\cdot}3 \text{ B.Th.U.} \end{aligned}$$

8. Wet Steam.—A steam boiler very rarely, if ever, generates dry saturated steam, but almost always wet steam. The amount of moisture present in the steam depends upon the type of boiler and the nature of the load. If the boiler is being forced and the steam liberating surface is too small, "priming" is liable to take place; when priming occurs, water in large quantities is carried bodily out of the boiler by the steam, and the result is not conducive to satisfactory working of the engines supplied with such steam. Priming should not occur in a boiler working under ordinary conditions, and should be carefully distinguished from the wet saturated steam which is delivered from most boilers.

Suppose that 1 pound of the wet steam contains x pound of dry steam, the remaining $(1 - x)$ pound being moisture mechanically suspended in the steam. The steam is said to have a *dryness fraction* "x," and it is obvious that of the original pound of water, only x, of a pound has been evaporated.

The total heat of evaporation of 1 pound of the wet steam is therefore,

H = sensible heat (h) + latent heat L × dryness fraction (x)

or $H = h + xL$ . . . . (7)

or $H = (t - 32) + xL$, where t is the temperature of saturation.

It should be noticed that when $x = 1$ (*i.e.* dry steam) (7) becomes the same as (1) Art. 5.

The external work done during evaporation will also be less than in the case of dry steam because the volume of each pound of wet steam will be less than if the steam were dry.

Using the same notation as in the preceding Art. 6, we have

External work done during evaporation (E′) = Pressure × change in volume.

Now the volume of 1 pound of steam whose dryness fraction is x will be $x \times V$ (neglecting the very small volume occupied by the $(1 - x)$ pound of water). Hence,

$$E' = P(xV - w) \text{ foot-pounds} \quad . \quad . \quad . \quad . \quad (8)$$

$$= \frac{P}{J}(xV - w) \text{ B.Th.U.} \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

A modern steam boiler working at its normal rate should produce steam containing not more than about 1·5 per cent. of moisture, or in other words, the dryness fraction of the steam should not be less than 0·985, *i.e.* the steam is 98·5 per cent. dry.

Example.—A boiler generates steam which is 98 per cent. dry at a pressure of 200 pounds per square inch ($t = 381{\cdot}5°$ F.). How many B.Th.U. must be supplied by the boiler per pound of steam produced if the temperature of the feed water is 60° F.?

Heat of formation

$$\begin{aligned}
&= \text{increase in sensible heat} + x \times \text{latent heat} \\
&= 381{\cdot}5 - 60 + 0{\cdot}98\,(1114 - 0{\cdot}7 \times 381{\cdot}5) \\
&= 381{\cdot}5 - 60 + 0{\cdot}98 \times 847 \\
&= 321{\cdot}5 + 830 \\
&= 1151{\cdot}5 \text{ B.Th.U.}
\end{aligned}$$

Note.—Less heat is required to produce wet steam than is required to produce the same weight of dry steam. In the above example, if the boiler generated dry saturated steam ($x = 1$), the heat required per pound of steam would be

$$\begin{aligned}
H &= 321{\cdot}5 + 847 \\
&= 1168{\cdot}5 \text{ B.Th.U.}
\end{aligned}$$

instead of 1151·5 B.Th.U.

9. Equivalent Evaporation from and at 212° F.—From the preceding Arts. 6, 7, and 8, it will be seen that the amount of heat required to produce 1 pound of steam depends upon the temperature of the feed water, upon the dryness of the steam if the steam is saturated, or upon the temperature of superheat if the steam is superheated. Suppose we wish to compare the performances of two different boilers; the actual evaporation per pound of fuel will be reduced in each case to the number of pounds of water which would be evaporated if the feed temperature were 212° F., and the steam formed at that temperature, *i.e.* to the equivalent evaporation from and at 212° F. The performances of the two boilers would then be directly comparable. An example will make this clear.

Example.—A boiler A generates 9 pounds of steam per pound of fuel, the steam being 98 per cent. dry, at a temperature of 350° F. from feed water at 50° F. Another boiler B generates 9 pounds of steam per pound of the same fuel, the steam being 95 per cent. dry at a temperature of 390° F. from feed water at 100° F. Which is the better boiler?

Total heat supplied by boiler A per pound of steam produced is

$$
\begin{aligned}
\text{increase in sensible heat} &+ x \times \text{latent heat} \\
&= 350 - 50 + 0{\cdot}98\,(1114 - 0{\cdot}7 \times 350) \\
&= 350 - 50 + 0{\cdot}98 \times 869 \\
&= 300 + 851{\cdot}6 \\
&= 1151{\cdot}6 \text{ B.Th.U.}
\end{aligned}
$$

$$
\left.\begin{array}{l}\text{Equivalent evaporation from} \\ \quad\text{and at } 212^\circ \text{ F.} \;.\;.\;.\;.\;.\end{array}\right\} = \frac{1151{\cdot}6}{966{\cdot}6} \times 9
$$

$$
= 10{\cdot}72 \text{ pounds per pound of fuel.}
$$

Total heat supplied by boiler B per pound of steam produced is

$$
\begin{aligned}
\text{increase in sensible heat} &+ x \times \text{latent heat} \\
&= 390 - 100 + 0{\cdot}95\,(1114 - 0{\cdot}7 \times 390) \\
&= 390 - 100 + 0{\cdot}95 \times 841 \\
&= 290 + 799 \\
&= 1089 \text{ B.Th.U.}
\end{aligned}
$$

$$\left.\begin{array}{l}\text{Equivalent evaporation from}\\ \quad\text{and at } 212^\circ \text{ F.} \ldots\ldots\end{array}\right\} = \frac{1089}{966 \cdot 6} \times 9$$

$$= 10 \cdot 13 \text{ pounds per pound of fuel}$$

The performance of boiler A is therefore better than that of B.

10. Factor of Evaporation.—The factor of evaporation is that factor by which the actual evaporation per pound of fuel under working conditions, must be multiplied by in order to obtain the equivalent evaporation from and at 212° F. In the above example, the factor for the boiler A is

$$\frac{\text{Heat supplied by the boiler per pound of steam produced}}{966 \cdot 6}$$

$$= \frac{1151 \cdot 6}{966 \cdot 6} = 1 \cdot 191$$

In symbols, the factor of evaporation is

$$\frac{(t + x\text{L}) - \text{temp. of feed-water}}{966 \cdot 6}$$

or if H is the total heat at the boiler working pressure as given in steam tables, reckoned from 32° F., and the steam being assumed to be dry saturated, it is

$$\frac{\text{H} + 32 - \text{temp. of feed-water}}{966 \cdot 6}$$

11. Efficiency of a Steam Boiler.—The efficiency of a steam boiler is represented by the expression

$$\frac{\text{Heat transferred to the steam}}{\text{Heat supplied by the fuel}}$$

Consider the example worked out in Art. 9. For each pound of steam generated by the boiler A, 1151·6 B.Th.U. are transferred from the fuel into the steam, and since 1 pound of the fuel generates 9 pounds of steam, we have

Heat transferred to the steam per pound of fuel = 1151·6 × 9 B.Th.U. If the calorific value of the fuel is 15,000 B.Th.U. per pound, the efficiency of the boiler will be

$$\frac{1151 \cdot 6 \times 9}{15{,}000} = 0 \cdot 691 \text{ or } 69 \cdot 1 \text{ per cent.}$$

The efficiency of the boiler B will be found similarly, namely

$$\frac{1089 \times 9}{15{,}000} = 0{\cdot}653 \text{ or } 65{\cdot}3 \text{ per cent.}$$

If we know the equivalent evaporation from and at 212° F., the efficiency of the boiler is easily found as follows :—

$$\text{Efficiency} = \frac{\text{Equivalent evaporation from and at } 212^\circ \text{ F} \times 966{\cdot}6}{\text{Calorific value of the fuel}}$$

For the boiler A this gives

$$\text{Efficiency} = \frac{10{\cdot}72 \times 966{\cdot}6}{15{,}000} = 0{\cdot}691 \text{ or } 69{\cdot}1 \text{ per cent. as before,}$$

and for the boiler B

$$\text{Efficiency} = \frac{10{\cdot}13 \times 966{\cdot}6}{15{,}000} = 0{\cdot}653 \text{ or } 65{\cdot}3 \text{ per cent. as before.}$$

12. Transmission of Heat across Boiler Heating Surfaces.—In a boiler furnace there are three methods by which heat is transmitted from the flame to the water in the boiler; namely, transmission by radiation, by convection, and by conduction. Heat waves from the flame are radiated to the furnace side of the plate through which their energy is transmitted by conduction to the water on the other side of the plate. The hot gases from the fire transmit heat to the plate by convection, the plate also conducting this heat through to the water. There are very few, if any, authentic records of experiments having for their object the analysis of the amount of heat transmitted by each of these three methods, and there is no simple formula which will give the heat transmitted for all conditions under which a furnace may be worked.

13. Transmission by Radiation.—According to Stefan the heat radiated from a black body is proportional to the fourth power of the absolute temperature, and the following formula may be used to estimate the number of British Thermal Units radiated per square foot of heating surface per hour from the flame in a boiler furnace,

$$H = \frac{16}{10^{10}} \times (T_1^4 - T_2^4) \text{ B.Th.U. per square foot per hour}$$

where T_1 is the temperature of the flame in degrees Fahrenheit

absolute, and T_2 is the temperature of the furnace side of the heating surface in degrees Fahrenheit absolute. From this formula it will be seen that the higher the temperature of the flame, the greater the quantity of heat transmitted by radiation; hence the importance of a high temperature of combustion.

Out of many experiments which prove that the heating surface exposed to the direct action of the incandescent gases is more efficient than that which is only exposed to hot gases which are not incandescent, the following may be mentioned.*

"*Messrs. Dewrance and Woods* † divided the water space between the smoke-box tube plate and the firebox tube plate in a locomotive boiler into six equal compartments by means of vertical partitions. It was found that the evaporation per square foot in the first compartment, *i.e.* the one next to the firebox, was equal to the evaporation per square foot of the firebox. In the second compartment the evaporation was one-third of that in the first compartment, and the evaporation in the remaining compartments was negligibly small.

"*Graham* ‡ divided a cylindrical boiler set in brickwork into four compartments each being open to the atmosphere, and found that 67·6 per cent. of the total water was evaporated in the first compartment, 18·2 per cent. in the second, 8·1 per cent. in the third, and 5·4 per cent. in the fourth compartment.

"*Wye Williams* || arranged six separate evaporating compartments round a common flue tube, the first compartment being quite separate from the other five, and its heating surface being directly exposed to the flame. He found that more water was evaporated from the first compartment, which was only one inch long, than from the last two feet of the tube.

"*Couche* divided a locomotive boiler into five compartments by means of tube plates. He found that the evaporation per square foot of heating surface was between two and three times greater from the firebox, than from the first compartment. He

* Prof. W. E. Dalby on "Heat Transmission," *Inst. Mech. Engineers*, Oct., 1909.

† *Trans. Inst. Nav. Arch.*, vol. 3, 1842.

‡ *Lit. Phil. Soc. Manchester*, vol. 15, 1860.

|| *Trans. Inst. Nav. Arch.*, vol. 3, 1862.

obtained an exceptionally high evaporation of 50 pounds per square foot per hour from the firebox part of the boiler with a draught of 3 inches of water in the smoke box. He also found that the effect of closing up half the tubes was to cause an increase in the amount of heat transmitted per square foot of heating surface."

14. Transmission by Convection and Conduction through the Metal Plates.—The simplest problem in the conduction of heat is that in which the opposite sides of a plate of indefinitely large area are maintained at two different constant temperatures. Heat is then conducted through the plate at a uniform rate, its amount being known from the simple formula,

$$H = k \times \frac{t_1 - t_2}{x} \text{ B.Th.U. per square foot per hour} \quad . \quad (1)$$

where t_1 and t_2 are the temperatures (° F.) of the two sides of the plate, x the thickness of the plate in inches, and k is the thermal conductivity of the plate, being the number of British Thermal Units passing per hour through a plate one inch thick when its sides are kept at a constant difference of temperature of 1° F. This constant has a value for wrought iron and mild steel of 450.

The difficulty attending the use of this formula lies in our uncertainty as to the temperature on the two sides of the plate. If we assume the temperature on the gas side of the plate to be, say, 1600° F., and on the water side 380° F., the above formula will give for a plate 0·5 inch thick.

$$H = 450 \times \frac{1600 - 380}{0{\cdot}5}$$

$$= 1{,}098{,}000 \text{ B.Th.U. per square foot per hour.}$$

This will give an equivalent evaporation from and at 212° F. of

$$\frac{1{,}098{,}000}{966} = 1130 \text{ pounds per square foot per hour.}$$

Now in actual practice an average evaporation of only about 5 pounds is obtained, corresponding to a transmission of only about 5000 B.Th.U. per square foot per hour, which would be obtained if the difference in temperature on the two sides of the

above plate was about 5·5° F. We are therefore forced to the conclusion that the temperature difference on the two sides of the plate is nothing like the difference in temperature between the hot gases and the water.

Rankine gave the empirical formula (deduced from the results of numerous experiments) for the heat transmitted per hour per square foot of surface,

$$H = \frac{(t_1 - t_2)^2}{a} \text{ B.Th.U. per square foot per hour.} \quad (2)$$

where a varies from 160 to 200, and t_1 and t_2 are the temperatures in ° F. of the hot gases and water respectively on the two sides of the plate.

Taking $a = 200$ and applying this formula to the above case, we find

$$H = \frac{(1600 - 380)^2}{200} = 7440 \text{ B.Th.U. per square foot per hour.}$$

If now we apply formula (1) to this result we can find the probable difference in temperature $(t_1 - t_2)$ required to transmit 7440 B.Th.U. per square foot per hour, for we have

$$H = 450 \times \frac{(t_1 - t_2)}{0{\cdot}5}$$

$$7440 = 900(t_1 - t_2)$$

$$t_1 - t_2 = \frac{7440}{900} = 8{\cdot}2° \text{ F.}$$

Now the temperature of the water side of the plate will not be very much higher than that of the water itself; suppose it to be, say, 420° F., this value agreeing fairly well with experimental observations. Then the temperature on the gas side of the plate will be, according to the above theory, $420 + 8 =$ say 430° F. Now the temperature of the hot gases has been assumed as 1600° F., hence we have a great drop in temperature between the gases in the flue and the heating surface, in this case $1600 - 430 = 1170°$ F.

The above conclusion is confirmed by the results of experiment which certainly show that the difference in temperature on the two sides of the heating surface is very small. Mr.

Hudson * found that the gas side of the plate forming the heating surface was never more than 36° hotter than the water side, while Sir John Durston † found the temperature on the gas side to be 68° higher than on the water side.

We are, therefore, justified in believing that the available temperature head across the plate is a very small percentage of that between the hot gases and the water.

15. On the existence of Gas and Water Films coating the Plate.—The great drop in temperature between the hot gases and the plate may be accounted for if we assume that there is a thin layer of gas which clings to the plate. Gases transmit heat chiefly by convection, being very bad conductors of heat, and if such a stationary gas film exists it will easily be seen that the thicker the film the greater will be the drop in temperature between the hot gases flowing over the heating surface and the metal plate. Experiment shows that this gas film does exist and its thickness is of the order of $\frac{1}{40}$ inch.

There is also considerable evidence to show that a thin film of water clings to the water side of the plate separating the bulk of water from the plate. Through this film, as in the case of the gas film, heat can only be transmitted by conduction, and water is a bad conductor of heat. The question, however, is complicated by the formation of steam bubbles.

Professor Dalby ‡ suggests that of the total temperature head, about 97 per cent. of the whole is required to overcome the resistance of the gas film, 1 per cent. to overcome the resistance of the plate and 2 per cent. to overcome the resistance of the water film. This means that the material of which the firebox and flue is constructed makes very little difference to the transmission of heat, and that iron or steel tubes and fireboxes are quite as good as copper ones and, further, that the thickness of the plate makes very little difference in the heat transmitted.

* *Engineer*, vol. 70, p. 523, 1890.

† *Trans. Inst. Nav. Arch.*, vol. 34, p. 130, 1893.

‡ "Heat Transmission," *Inst. Mech. Engineers*, Oct., 1909.

16. Effect of High Speed upon Conduction.—The formulæ given in Art. 14 take no account of the speed with which the hot gases sweep over the heating surface, nor of the density of those gases. The greater the speed of the gases, the thinner will be the gas film adhering to the plate forming the heating surface, and the higher will be the temperature on the gas side of the plate resulting in an increased temperature head across the plate and consequently in an increased amount of heat transmitted. To some extent also, the same holds on the water side of the heating surface.

Professor Osborne Reynolds in 1874 deduced from theoretical considerations that the amount of heat transmitted was a linear function of the speed of the fluids. He gave the following formula for the amount of heat passing from the gases to the heating surface

$$Q = (A_1 + B_1\rho_1\mu_1)(T - \theta) \quad . \quad . \quad . \quad (1)$$

where Q = B.Th.U. transmitted per square foot per second,
ρ_1 = density of the gases in pounds per cubic foot,
μ_1 = velocity of the gases in feet per second,
T = temperature of the gases in °F.,
θ = mean temperature of the heating surface,
A_1 and B_1 = experimental constants.

In 1897, Dr. T. E. Stanton showed that the above law held for water on opposite sides of a metal plate;* the law may be written

$$Q = (A_1 + B_1\rho_1\mu_1)(T - \theta) = (A_2 + B_2\rho_2\mu_2)(\theta - t) \quad (2)$$

where ρ_2 = density of water in pounds per cubic foot,
μ_2 = velocity of the water in feet per second,
t = temperature of the water in °F.,
A_2 and B_2 = constants.

The above law has since been verified by many notable experimenters, but for a detailed account of the researches which have been made on "Heat Transmission," the reader is referred to the paper by Professor Dalby † already mentioned.

* *Trans. Royal Society*, vol. cxcix, 1897, pp. 67–88.

† *Proc. Inst. Mech. Engineers*, Oct., 1909.

Professor J. T. Nicolson in 1909 gave the above formula for the rate of heat transmission in a somewhat elaborate form.* He found, as the result of experiment, that the rate of heat transmission depended not only on the product of speed and density, but also to some extent on the average value of the gas and wall temperatures, the hydraulic mean depth of the tubes or flues through which the gases passed, and upon the nature of the metal surface in contact with the gas. According to Nicolson,

$$Q = \left[\frac{\phi}{200} + \frac{\sqrt{\phi}}{40}\left(1 + \frac{1}{m_1}\right)\rho_1\mu_1\right](T - \theta) \quad . \quad . \quad (3)$$

or

$$Q = \left[\frac{\phi}{200} + \frac{\sqrt{\phi}}{40}\left(1 + \frac{1}{m_1}\right)\frac{w_1}{a_1}\right](T - \theta) \quad . \quad . \quad (4)$$

where Q = B.Th.U. transmitted per hour per square foot of heating surface,

T = temperature of gas flowing along the flue in °F.,

θ = temperature of the metal wall in °F.,

$\phi = \frac{1}{2}(T + \theta)$ = mean film temperature in °F.,

ρ_1 = density of gas in pounds per cubic foot,

μ_1 = speed of gas in feet per second,

w_1 = pounds of gas flowing per second,

a_1 = area of flue in square feet,

m_1 = hydraulic mean depth of the flue in inches,

$= \dfrac{\text{area of flue in square inches}}{\text{perimeter of flue in inches}}$.

Lack of space forbids a detailed analysis of his paper, but it will be instructive to take from it the following example:—

Lancashire boiler flue, 36 inches diameter; coal burnt on a grate 20 square feet in area, 400 pounds per hour; air supplied per pound of coal, 24 pounds. Temperature of the gases leaving the fire 2200° F., temperature of the gases at the end of the flue 900° F., steam temperature 350° F.

* *Trans. Jun. Inst. of Engineers*, Feb., 1909, and *Proc. Inst. of Mech. Engineers*, Oct., 1909.

Here the average temperature of the gases is

$$T = \frac{2200 + 900}{2} = 1550^\circ \text{ F.}$$

Mean film temperature

$$\phi = \frac{1550 + 350}{2} = 950^\circ \text{ F.}$$

Gas flow per second, $w_1 = \dfrac{400 \times 25}{3600}$ pounds

and since $a_1 = \frac{\pi}{4} \times (3)^2 = 7$ square feet,

$$\frac{w_1}{a_1} = \frac{400 \times 25}{3600 \times 7} = 0{\cdot}4.$$

Also since the hydraulic mean depth

$$m_1 = \frac{7 \times 144}{\pi \times 36} = 9 \text{ inches,}$$

$$1 + \frac{1}{m_1} = 1 + \frac{1}{9} = 1{\cdot}11.$$

$$\begin{aligned} \text{Therefore } Q &= \left[\frac{950}{200} + \sqrt{\frac{950}{40}} \times 1{\cdot}11 \times 0{\cdot}4\right](1550 - 350) \\ &= (4{\cdot}75 + 0{\cdot}77 \times 1{\cdot}11 \times 0{\cdot}4)1200 \\ &= (4{\cdot}75 + 0{\cdot}342)1200 \\ &= 5{\cdot}092 \times 1200 \\ &= 6100 \text{ B.Th.U. per square foot per hour.} \end{aligned}$$

The figure 5·092 B.Th.U. per square foot per hour per degree average difference of temperature between the gases and water, compares very well with the figures deduced from practice by Mr. Michael Longridge (Manchester Association of Engineers, 1890), who gives for such a case 4·9 to 5·5 B.Th.U. per square foot per hour per degree difference of temperature.

Comparison of the Results obtained from the use of Rankine's and Nicolson's Formulæ.—The formula recommended by Rankine and commonly used in heat transmission calculations for steam boilers is, as already mentioned in Art. 14,

$$Q = \frac{(T - t)^2}{160 \text{ to } 200}$$

where Q = heat transmitted per square foot per hour (B.Th.U.)
T = average gas temperature in °F.
t = water or steam temperature in °F.

This may be written :—

$$Q = H(T - t) \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

where H = heat transmitted (in B.Th.U.) per square foot per hour per degree difference of temperature.

Nicolson's formula is

$$Q = \left[\frac{\phi}{200} + \frac{\sqrt{\phi}}{40}\left(1 + \frac{1}{m_1}\right)\frac{w_1}{a_1}\right](T - t)$$

so that in it

$$H = \left[\frac{\phi}{200} + \frac{\sqrt{\phi}}{40}\left(1 + \frac{1}{m_1}\right)\frac{w_1}{a_1}\right] \quad . \quad . \quad . \quad . \quad (6)$$

Here $\phi = \frac{1}{2}(T + t)$

$m_1 = \frac{d}{4}$ = hydraulic mean depth of the flue.

w_1 = weight of gases passing through the flue per second (in pounds).

a_1 = sectional area of the flue in square feet.

We may calculate the values of H in each of the expressions (5) and (6), and tabulate them for comparison in a few practical cases.*

Rankine's formula $\frac{(T - t)^2}{200}$ *applied to a Lancashire Boiler.*

$t = 350°$ F.

T =	2050° F.	1550° F.	1050° F.	550° F.
$T - t$ =	1700°	1200°	700°	200°
$H = \frac{T - t}{200}$ =	8·5	6	3·5	1

* These examples are taken from Prof. Nicolson's paper in *Trans. Junior Inst. of Engineers*, vol. xix., Feb., 1909.

Nicolson's formula applied to a Lancashire Boiler.

$$m_1 = \frac{d}{4} = \frac{40}{4} = 10 \text{ inches.} \quad \therefore\ 1 + \frac{1}{m_1} = 1{\cdot}1$$

Taking $\rho_1 = 0{\cdot}02$ and $\mu_1 = 25$ we have

$$\left(1 + \frac{1}{m_1}\right)\rho_1\mu_1 = 0{\cdot}55$$

$t = 350°$ F.

T	=	2050° F.	1550° F.	1050° F.	550° F.
$\phi = \frac{1}{2}(\mathrm{T} + t)$	=	1200°	950°	700°	450°
$\frac{\phi}{200}$	=	6	4·75	3·5	2·25
$\frac{\sqrt{\phi}}{40}$	=	0·875	0·78	0·673	0·545
$\frac{\sqrt{\phi}}{40}\left(1 + \frac{1}{m_1}\right)\rho_1\mu_1$	=	0·48	0·43	0·37	0·30
H_1	=	6·48	5·18	3·87	2·55

As the ordinary values of the mean gas temperature in Lancashire boilers range from 1100° F. to 1300° F., it will be seen that Rankine's formula agrees with Nicolson's over that range.

Rankine's formula $\frac{(\mathrm{T} - t)^2}{160}$ *applied to Locomotive Boilers, moderate draught.*

$t = 350°$ F. as before.

T	=	2050° F.	1550° F.	1050° F.	550° F.
$\mathrm{T} - t$	=	1700°	1200°	700°	200°
$\mathrm{H} = \frac{\mathrm{T} - t}{160}$	=	10·63	7·5	4·37	1·25

Nicolson's formula applied to Locomotive Boilers, moderate draught.

$$m_1 = \frac{d}{4} = \frac{1{\cdot}75}{4} = 0{\cdot}44 \text{ inches.} \quad \therefore 1 + \frac{1}{m_1} = 3{\cdot}3.$$

Taking $\rho_1 = 0{\cdot}02$ and $\mu_1 = 50$ feet per second, we have

$$\left(1 + \frac{1}{m_1}\right)\rho_1\mu_1 = 3{\cdot}3.$$

$t = 350°$ F. as before.

T	=	2050° F.	1550° F.	1050° F.	550° F.
$\phi = \frac{1}{2}(\mathrm{T} + t)$	=	1200°	950°	700°	450°
$\frac{\phi}{200}$	=	6	4·75	3·5	2·25
$\frac{\sqrt{\phi}}{40}$	=	0·875	0·78	0·673	0·545
$\frac{\sqrt{\phi}}{40}\left(1 + \frac{1}{m_1}\right)\rho_1\mu_1$	=	2·9	2·57	2·23	1·8
H_1	=	8·9	7·32	5·73	4·05

Thus, in the case of locomotive boilers working with moderate draught and a gas speed of 50 feet per second, by a suitable, but quite arbitrary, change of the constant a in the formula $\frac{(\mathrm{T} - t)^2}{a}$ we have again obtained agreement between the two formulæ, over the usual range of gas temperature which occurs (say from 1300° F. to 1100° F.).

Rankine's formula $\frac{(T-t)^2}{120}$ *applied to Locomotive Boilers with high draught.*

$t = 350°$ F. as before.

T	=	2050° F.	1550° F.	1050° F.	550° F.
$T - t$	=	1700°	1200°	700°	200°
$H = \frac{T-t}{120}$	=	14·2	10	5·9	1·7

Nicolson's formula applied to Locomotive Boilers with high draught.

$$1 + \frac{1}{m_1} = 3{\cdot}3 \text{ as before.}$$

Taking $\rho_1 = 0{\cdot}02$ and $\mu_1 = 150$ feet per second, we have

$$\left(1 + \frac{1}{m_1}\right)\rho_1\mu_1 = 10.$$

$t = 350°$ F. as before.

T	=	2050° F.	1550° F.	1050° F.	550° F.
$\phi = \frac{1}{2}(T + t)$	=	1200°	950°	700°	450°
$\frac{\phi}{200}$	=	6	4·75	3·5	2·25
$\frac{\sqrt{\phi}}{40}$	=	0·875	0·78	0·673	0·545
$\frac{\sqrt{\phi}}{40}\left(1 + \frac{1}{m_1}\right)\rho_1\mu_1$	=	8·75	7·8	6·73	5·45
H_1	=	14·73	12·55	10·25	7·70

Here we see that even with a further arbitrary reduction of the constant *a* to 120 (as recommended by Prof. R. H. Smith) we are only able to obtain agreement between the two formulæ

for such high values of the average gas temperature as to be almost beyond the range of actual practice. For the usually occurring values of this temperature (say from 1500° F. to 1300° F.) the old formula, due to Rankine, gives rates of heat transmission much lower than the correct amount.

From the above figures it will be seen that for boilers working with ordinary draught, and therefore low gas speeds, Rankine's formula gives results which are practically in agreement with those obtained in practice, and this, together with the fact that increased draught results in a higher furnace temperature (and therefore a higher mean gas temperature), and an increased rate of combustion, is doubtless the reason why the inaccuracy of Rankine's formula was not detected until recently in the case of boilers working with a high forced draught.

17. Method of Creating High Gas Speeds.—In order to produce the high gas speed recommended by Prof. Nicolson (up to 250 feet per second) a powerful draught is necessary. This, however, does not mean an abnormally high rate of combustion, for no matter how great the difference of air pressure between the furnace and chimney may be, that between furnace and ashpit need not, and preferably should not, be greater than what is commonly employed when firing under "natural draught." It may, for example, be found that, for a certain type of boiler a vacuum of 20 inches of water at the fan suction gives, on the whole, the most economical results, whilst, at the same time, it will appear that the rate of firing should be limited to 25 or 30 pounds of coal per hour per square foot of grate area (see Art. 18).

In such a case almost the whole of the 20 inches of draught, produced by the fan, is spent upon drawing the gases through the flues at high speed, in the evaporating and economising portions of the boiler; whereas, with the systems hitherto used, the greatest resistance to the passage of the gas has always been offered by the fire itself, and a high blast meant a thick fire and a high rate of firing which resulted (when the draught was high enough) in low efficiency (see Art. 9, Chap. IV.) on account of incomplete combustion.

18. The most Efficient Rate of Combustion.*—In all boilers the two chief sources of loss are furnace loss and chimney loss. *Furnace loss* is that fraction of the total heat in the coal which goes to waste on account of incomplete combustion, or the non-combustion of parts of its constituents. *Chimney loss* is that fraction of the total heat in the coal which goes to waste by reason of the products of combustion leaving the boiler at temperatures higher than that of the entering feed (Art. 9, Chap. II.).

Furnace Loss.—The chief source of loss with high draughts is the blowing out of the fire of coal dust and small coal, which is carried right through the flues without being burned, and goes away up the chimney. At low rates only very fine dust escapes in this way and the action is possibly unimportant; but with the fierce blast of a locomotive running at full speed, there is a constant rain of quite large pieces of coal from the top of the funnel, and by far the greatest proportion of all the loss occurred in the furnace is due to this cause.† Professor Nicolson finds that the combined loss of heat due to imperfect combustion and coal blown away amounts to 50 F., B.Th.U. per pound of coal and that the heat actually generated in the fire per pound of coal is

$$Q_0 = Q - 50\,F. \quad \ldots\ldots \quad (1)$$

where Q = calorific value of the coal in B.Th.U. per pound,

F = rate of firing in pounds of coal per square foot of grate per hour.

Chimney Loss.—By Art. 9, Chap. II. the chimney loss is

$$W \times s\,(t_1 - t_2) \text{ B.Th.U. per pound of coal.}$$

where W = weight in pounds of the flue gases per pound of coal burned,

s = mean specific heat of the flue gases,

t_1 = temperature (°F.) of the flue gases leaving the boiler,

t_2 = temperature (°F.) of air in boiler house.

* For the substance of this Article the Author is indebted to Prof. Nicolson's paper on "Boiler Economics."

† See papers by F. J. Brislee and L. H. Fry in *Proc. Inst. Mech. Engineers*, 1908, p. 237 and p. 269.

The chimney temperature t_1 may be calculated from the formula *

$$\frac{s}{a} = c \log_e \frac{T_1 - t}{t_1 - t} \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

where s = area of heating surface in square feet,
a = cross-sectional area of flue in square feet,
T_1 = furnace temperature,
t_1 = chimney temperature,
t = steam temperature.

(2) may be written,

$$\epsilon^{\frac{s}{ca}} = \frac{T_1 - t}{t_1 - t}$$

or

$$t_1 = t(1 - \epsilon^{\frac{s}{ca}}) + T_1 e^{-\frac{s}{ca}} = A + BT_1 \quad . \quad . \quad (3)$$

from which it is seen that for any given boiler the chimney temperature is a linear function of the furnace temperature.

Furnace Temperature.—The heat generated in the fire per square foot of grate per hour is $Q_0 \times F$. This heat is disposed of in two ways—

(1) By radiation from the fire surface to the furnace plates.

(2) By heat communicated to the products of combustion, their temperature being thereby raised from t_2 to T_1.

By Stefan and Boltzmann's law of radiation the quantity of heat radiated per hour per square foot of fire surface is

$$R = 1600 \left(\frac{\tau_0}{1000}\right)^4 \text{B.Th.U.} \quad . \quad . \quad . \quad . \quad (4)$$

where $\tau_0 = T_1 + 461$, the absolute temperature of the fire surface. The heat received by the furnace gases per square foot of grate per hour is

$$W \times s \times F\,(T_1 - t_2) \text{ B.Th.U.}$$

* For proof of this formula see "Boiler Economics and High Gas Speeds," by Prof. J. T. Nicolson, *Trans. Inst. of Engineers and Shipbuilders in Scotland*, 1911.

Therefore, the heat equation for one square foot of grate may be written

$$(Q - 50F)F = 1600\left(\frac{\tau_0}{1000}\right)^4 + W \times s \times F(\tau_0 - t_0) \quad (5)$$

where t_0 = absolute temperature of air supply $= t_2 + 461$.

From the study of many boiler trials it appears that the amount of air supplied per pound of coal with good firing is

$$A = \left(\frac{300}{F} + 9\right) \text{ pounds per pound of coal.}$$

Hence the weight of flue gases per pound of coal is A + 1 or

$$W = \frac{300}{F} + 10 \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

Substituting (6) in (5) the heat equation reduces to

$$\left(\frac{\tau_0}{1000}\right)^4 + (46{\cdot}9 + 1563F)\left(\frac{\tau_0}{1000}\right)$$
$$= \frac{(Q - 50F)}{1600}F + 24{\cdot}4 + 0{\cdot}812\,F. \quad . \quad . \quad . \quad (7)$$

From (7) the absolute furnace temperature τ_0, and therefore T_1, may be calculated for various values of F. If this be done and a curve be plotted connecting T_1 and F, it will be seen that the furnace temperature, T_1, as thus determined, rises very rapidly at first as F increases, reaches a maximum for F = 80, and then begins to fall. One ought to expect an increase of the furnace temperature with an increased rate of combustion, because less air is supplied per pound of coal, and there is, consequently, a smaller weight of products to be heated up. Recalling, however, the fact that the heat actually generated in the fire per pound of coal gets less and less, owing to imperfect combustion and coal blown away, it will readily appear that this source of loss will presently overtake the gain due to diminished air supply, the furnace temperature curve will rise less and less steeply, and will finally attain a maximum and afterwards fall.

The chimney temperature t_1 may now be found for the various values of τ_0 (or T_1) by using equation (3), and hence the chimney loss. By adding the chimney loss to the furnace loss the whole loss of heat in the boiler due to these two causes together for various rates of firing may be found. If this be

done it will be found that the most efficient rate of firing occurs at between 25 and 30 pounds of coal per hour per square foot of grate area. For smaller values, the combustion is more perfect, but the larger chimney loss, due to the large amount of air supplied, more than makes up for this. For higher rates the chimney loss gets less, but imperfect combustion and coal blown away out of the fire does away with the advantage.

Professor Nicolson's proposals for the better design of steam boilers may be briefly summed up as follows:—

(1) To obtain a small heating surface there must be a high gas speed and narrow flues. This does not involve high rates of combustion as already mentioned in the preceding Art. 17.

(2) To pay for the fan power required to produce the high gas speed, there must be a counter-current economiser (p. 252) and low chimney temperature.

(3) To avoid external corrosion from low temperature gases some protection to the tubes was necessary. The most notable result of Professor Nicolson's boiler experiments was that this protection was found to come naturally by the formation of a thin coating of soot. With low gas speeds this would prove a great hindrance to heat transmission, but with the high speeds advocated, the extra heat resistance was almost negligible.

(4) For efficient combustion, either slow combustion on a large grate, or, if only a small grate and consequent hard firing be adopted, a properly designed reverberating furnace must be added in order to complete the combustion of the coal dust blown out of the fire.

Notwithstanding the large number of researches and papers bearing upon this subject there is a general absence of complete data regarding the actual phenomena occurring in a steam-boiler when working under ordinary conditions of practice. Boiler engineers are loth to revolutionise their methods of design until more definite evidence in favour of high gas speeds is forthcoming, although Professor Nicolson * is showing the way.

* See the following papers by Prof. Nicolson, "Laws of Heat Transmission in Steam Boilers," *Inst. Junior Engineer*, 1908, and "Boiler Economics, and the use of High Gas Speeds," *Inst. of Eng. and Shipbuilders in Scotland*, 1911.

The modern steam-boiler has reached its present state of perfection by a process of evolution, and each type may be said to be the survival of the fittest. The new method of design involving the use of high gas speeds would result in a very much smaller heating surface, and therefore in a smaller boiler having the same evaporative power. Such a boiler, if found in actual practice to be reliable, would have an enormous advantage over all existing types, especially for marine work where dead weight and space are such important factors. There is no reason to suppose that the new type of boiler would have a higher efficiency than present types because the losses mentioned in Art. 1, Chap. III. will still exist; the practical results obtained in this direction will be awaited with interest by all engineers concerned in the subject.

Examples on Chapter V.

1. Calculate:

(*a*) The total heat of 1 pound of dry saturated steam at 160 pounds per square inch absolute ($t = 363{\cdot}4°$ F.).

(*b*) The total heat of 1 pound of saturated steam of dryness fraction 0·95 at 160 pounds per square inch absolute ($t = 363{\cdot}4°$ F.).

(*c*) The total heat of 1 pound of steam at the above pressure but superheated to a temperature of 550° F.

Ans. (*a*) 1191 B.Th.U.; (*b*) 1148 B.Th.U.; (*c*) 1280 B.Th.U.

2. A boiler generates steam which is 98 per cent. dry at a pressure of 170 pounds per square inch absolute ($t = 368{\cdot}3$). How many B.Th.U. must be supplied by the boiler per pound of steam produced if the temperature of the feed water is 65° F.? What is the factor of evaporation?

Ans. 1142·3 B.Th.U.; 1·18.

3. A boiler A generates 8·7 pounds of saturated steam per pound of fuel, the steam being 98·5 per cent. dry and at a temperature of 380° F; the temperature of the feed water being 50° F. Another boiler, B, delivers 8·7 pounds of superheated steam per pound of the same fuel, at a pressure of 150 pounds per square inch absolute ($t = 358{\cdot}3$) and at a temperature of 500° F., the temperature of the feed water being 85° F. Calculate in each case (*a*) the equivalent evaporation per pound of fuel from and at 212° F.; (*b*) the thermal

efficiency of each boiler. Calorific value of the fuel = 13,500 B.Th.U. per pound.

Ans. Boiler A (*a*) 10·48 pounds; (*b*) 75·1 per cent. *Boiler B* (*a*) 10·84 pounds; (*b*) 77·6 per cent.

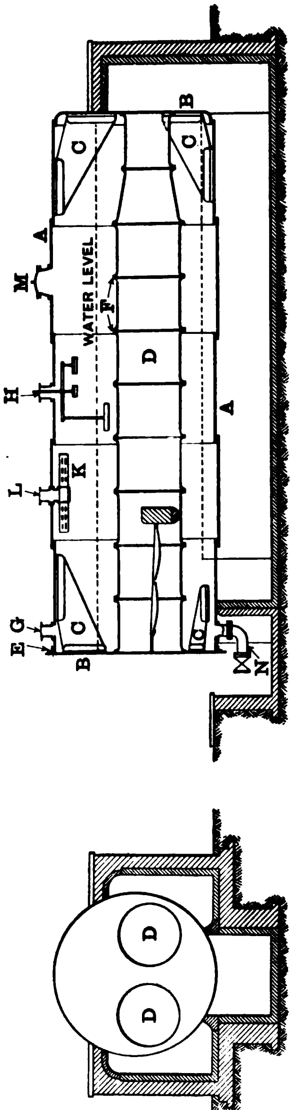
4. A Lancashire boiler furnace tube is 3 feet diameter. The coal burned per hour, 440 pounds; air supplied per pound of coal, 20 pounds; mean temperature of the gases in the tube, 1600° F.; steam temperature, 380° F. Calculate the quantity of heat transmitted per square foot of heating surface per hour, (*a*) according to Rankine's formula $Q = \frac{(T - t)^2}{200}$; (*b*) according to Nicolson's formula (Art. 16).

Ans. (*a*) 7442 B.Th.U.; (*b*) 6425 B.Th.U.

CHAPTER VI

CYLINDRICAL BOILERS OF THE SMOKE-TUBE TYPE

1. **The Lancashire Boiler.**—For steady and reliable working under the conditions obtaining in the majority of textile mills and factories in this country, the Lancashire boiler has proved itself to be eminently satisfactory. The construction of this type of boiler is shown in Fig. 35. It consists of a cylindrical shell A with flat ends B, stiffened with gusset stays C, and fitted with two furnace tubes D passing longitudinally from the front end plate to the back end plate. A furnace described on p. 56 is arranged at the front end of each furnace tube as shown. The outside shell A consists of a number of steel plates riveted together, the front end plate being attached to it by means of an external angle iron E, whilst the back end plate is flanged over and fits inside the shell to which it is riveted. The furnace tubes D are riveted to the two end plates and must be made flexible in order to allow for the difference of the temperature between them and the outer shell when the boiler is under steam. In round figures, considering a Lancashire boiler 30 feet long, if the furnace tubes could expand independently they would become about $\frac{1}{2}$ inch longer than the outer shell when under steam, this difference in length resulting solely from the fact that their mean temperature is considerably higher than the mean temperature of the shell. Now the flat end plates B are securely tied to the outer shell by means of the gusset stays C, and if the furnace tubes are as rigid as the outer shell they will be subjected to a very high longitudinal compressive stress. In endeavouring to relieve themselves of this stress the tubes become longer, and since the end plates



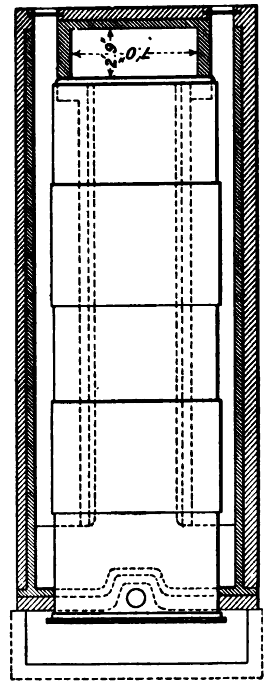


Fig. 35.—Lancashire boiler and setting.

are more or less rigid the tubes can only increase in length by bending. The joints F in the tubes should be flexible in order to allow for this expansion and contraction due to temperature differences. In some cases the necessary flexibility is secured by employing corrugated furnace tubes similar to those shown in the boilers of Figs. 42 and 49A. The boiler is usually fitted with a dead-weight safety valve at G and a combined lever safety valve and low-water alarm at H, which in addition to blowing off steam when the pressure is too high, blows a whistle when the water level is too low.

In order that the steam may be taken from the boiler as dry as possible an *anti-priming pipe* K is fitted. This pipe which is of suitable diameter and several feet long has its ends plugged up, and a number of holes drilled in it along its length in the upper half of its circumference. No steam can, therefore, be taken from the boiler through the stop valve at L except through these holes in the upper half of the pipe, *i.e.*, the part of the pipe remote from the water surface. The manhole M is fitted with a steam-tight door of elliptical shape for cleaning and inspection purposes. A blow-off cock N is fitted in the lowest and coolest part of the boiler in order to empty the boiler or blow out sediment, etc., when required.

The diameter of the shell varies from 6 feet 6 inches to 9 feet depending upon the rate of evaporation required, and the length is usually made about four times the diameter, a very common size being 8 feet diameter and 30 feet long constructed for a working pressure of from 160 to 180 pounds per square inch gauge (Table I, p. 4). The diameter of the furnace tubes is usually about 9 inches less than the radius of the shell.

The boiler is installed in a brickwork setting, a common arrangement being shown in Fig. 35. The hot gases pass along the furnace tubes from the furnace to the back, then down the end of the boiler shell (*i.e.* the downtake) and along the bottom flue from back to front. Here they divide into two streams which flow from front to back along the two side flues on their way to the chimney. When the boiler is laid off for cleaning or inspection it is necessary for a man to move along the side and bottom flues, and in order that he may conveniently

do so, the minimum width of the narrowest part of the side flues, *i.e.* the width measured at the horizontal diameter of the boiler shell, should be about 1 foot, while the least depth of the bottom flue should be about 2 feet; the usual width of the bottom flue is about half the diameter of the shell. Fig. 35 shows the brickwork setting for a Lancashire boiler, by Messrs. Galloways, Ltd., of 9 feet diameter and 30 feet long; the least width of the side flues is 1 foot, and the depth of the bottom flue at the front end 3 feet 6 inches.

If the brickwork setting is built so that the tops of the side flues and downtake are higher than the water level, the extra area of shell plate exposed to the hot gases will not contribute anything to the heating surface of the boiler because it is not covered with water on the inside. Such an arrangement would be a disadvantage because that portion of the shell exposed to the gases would be liable to get overheated, resulting in an increase of the temperature stresses set up in the boiler. For these reasons the tops of the side flues and downtake should not extend above the normal water level in the boiler.

Lime mortar should not be used for the brickwork in contact with the boiler shell because it is liable to absorb moisture when the boiler is not at work, and then by chemical reactions causes external corrosion of the plates; fireclay, which is free from this disadvantage, should be used in preference. The inside of all the flues should be lined with firebrick 4½ inches thick (*i.e.* one brick thick) as shown, and to facilitate draining of the boiler by the blow-off cock, it should be set with a fall towards the front end of about ½ inch for every 10 feet length of boiler. The blow-off cock (N, Fig. 35) should be quite free from the brickwork and be easily accessible for inspection as shown, whilst the width of the hearth pit should be sufficient to allow plenty of room for the blow-off pipe, at least 3 feet being necessary for this purpose.

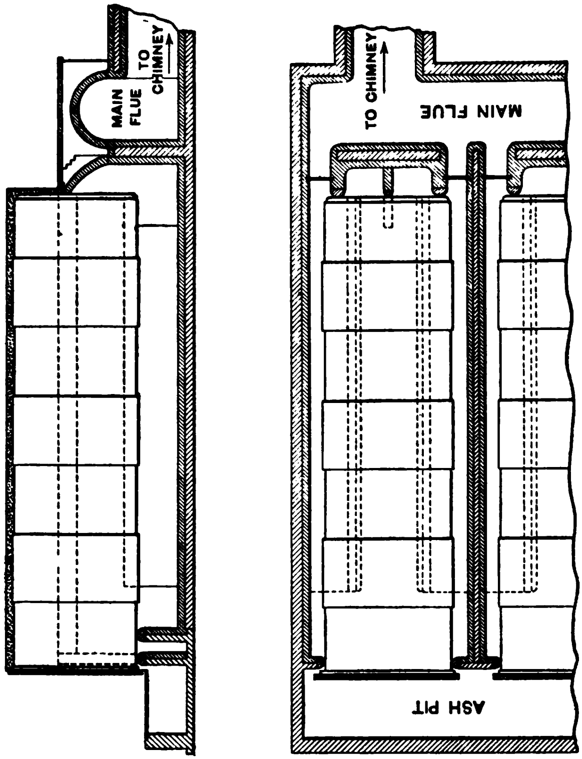
Although the setting shown in Fig. 35 is commonly used, it has two defects, one of which is in the downtake. Referring to the plan it will be seen that the section of the downtake is 7 feet by 2 feet 9 inches. At the exit of the gases from the two furnace tubes there is a tendency for the two streams

mixing together to cause a baffling of the draught, which may give rise to an unpleasant vibration of the furnace doors. This defect may be obviated by building a partition wall midway between the two furnace tubes which divides the downtake into two compartments and projects about 2 or 3 feet into the bottom flue as shown in Fig. 36.

The other defect is the large area of plate obscured by the brickwork in contact with it which makes complete inspection difficult. The boiler setting of Messrs. Poulton and Son, of Reading, is designed to obviate this defect and renders complete inspection possible. Fig. 36 shows their setting applied to a battery of Lancashire boilers, and it will be noticed that no portion of the surface of the plate is concealed, line contact between the plates and setting being obtained by using curved firebrick seating blocks.

Poulton's latest method of setting applied to a Cornish boiler is shown in the cross section, Fig. 37. This method is intended to prevent loss of heat by radiation, an air jacket surrounding the inside lining of firebrick. It is doubtful whether these air spaces are of any real advantage in reducing radiation losses. The United States Geological Survey have recently carried out experiments on this problem the results of which, as reported in Bulletin 8 of the U.S. Bureau of Mines, are expressed as follows:

"In furnace construction a solid wall is a better heat insulator than a wall of the same thickness containing an air space. This statement is particularly true if the air space is close to the furnace side of the wall, and if the furnace is operated at a high temperature. If it is advisable in furnace construction to build the wall in two parts, so as to prevent cracks being formed by the expansion of the brickwork on the furnace side of the walls, it is preferable to fill the space between the two walls with some 'solid' (not firm, but loose) insulating material. Any such easily obtainable materials as ash, crushed brick, or sand offer higher resistance to heat flow through the walls than an air space. Furthermore, any such loose material by its plasticity reduces air leakage, which is an important feature deserving consideration."



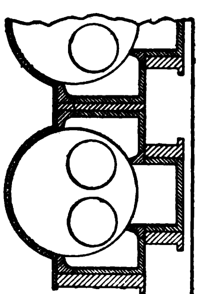


FIG. 36.—Lancashire boiler setting, by Messrs. Poulton and Sons.

This is in contradiction to the general belief that, as air is a poor heat conductor, air spaces in furnace walls will prevent or reduce losses. In explanation, it is said that air conducts heat very slowly, but that heat is radiated from a hot body through air very readily. The quantity of heat that passes through the solid part of the wall (by conduction) depends upon the difference in temperature between the hotter and cooler portions, while the heat that passes through the air space (by radiation) depends upon the difference between the fourth powers of

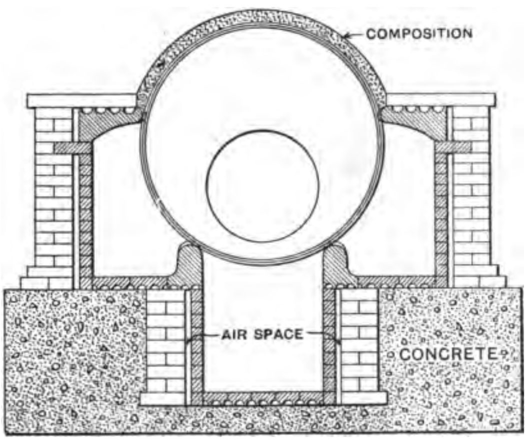


FIG. 37.—Poulton's latest setting.

the absolute temperatures (see p. 124) of the surface surrounding the air space.

The usual proportions of Lancashire boilers given above allow about 25 square feet of heating surface to each square foot of grate area (Table I., p. 4) and the boiler with ordinary chimney draught will evaporate from 160 pounds of water per square foot of grate area per hour to about 200 pounds in the larger sizes. When properly proportioned, the spaces between tubes, and between flues and boiler shell inside and outside, are such as to render the boiler easy to inspect, clean and keep in repair. Owing to its large contents of water and large steam

space, it is able to cope with considerable fluctuations of the load without any very great variation in the steam pressure. Further it is possible to use with a Lancashire boiler impure feed-water which in a water-tube or multitubular boiler with

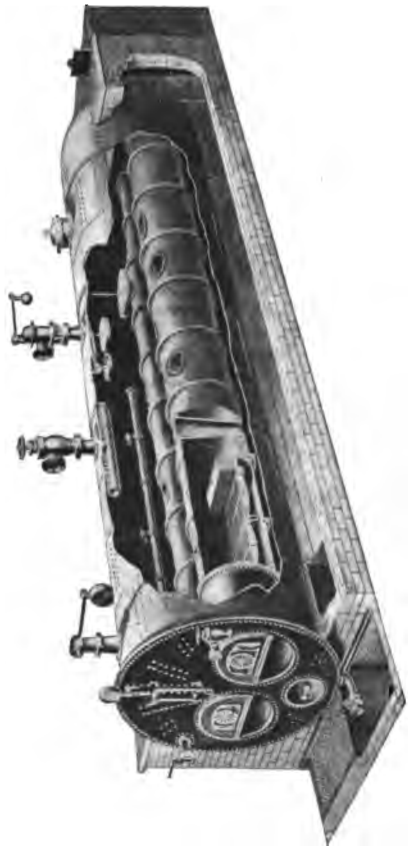


FIG. 38.—Lancashire boiler (Messrs. Galloways, Ltd.).

its small water contents would quickly cause an excessive deposit of sediment.

These advantages are, however, accompanied by certain disadvantages which render the boiler unsuitable in many

cases. The water circulation, especially between the flues and bottom of the shell is very sluggish, and this together with the large amount of water which it contains as compared with the water-tube boiler makes it impossible on an emergency to raise steam quickly from cold water. Any attempt to do this produces large stresses in the seams and joints, due to local difference of temperature in the various parts of the shell. Again, as compared with the water-tube boiler it cannot be easily divided into component parts, so that its erection in countries where transport is difficult is impracticable.

The Lancashire boiler is an economical steam generator, and where there is room for its installation it is regarded almost as the standard type in this country for general mill and factory work.

The Lancashire boiler is frequently constructed with Galloway cross tubes fitted in the two furnace tubes. The introduction of these cross tubes increases the heating surface and in addition it is claimed that the water circulation is improved. Fig. 38 shows a Lancashire boiler fitted with these cross tubes as manufactured by Messrs. Galloways, Limited.

Multi-flued Boilers.—If the diameter of the boiler is large enough, it becomes feasible to employ three instead of two furnace tubes as shown diagrammatically in Fig. 39, so as to obtain more grate area and a larger heating surface. But if the furnaces are fired with coal, the two side furnaces in a boiler of small diameter become too high above the floor level to be easily stoked, and the steam space above the water is much reduced. For these reasons, together with the difficulty of inspection and cleaning, the three-flued boiler has not been extensively employed. When gas fuel is used, the same objection does not apply, and in order to utilise the waste gases issuing from coke ovens and blast furnaces, Messrs. Galloways have introduced the Five-

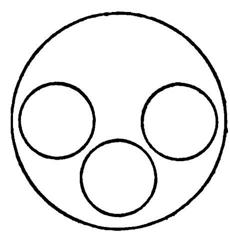


FIG. 39.—Three-flued boiler.

flued Boiler shown in Fig. 40. The standard boiler is 8 feet 6 inches diameter, 30 feet long, and the evaporation obtained in actual work varies from 8000 pounds to 10,000 pounds per hour, depending upon the number of ovens or furnaces in operation and the calorific value of the coal used. It will be noticed that the special feature consists in making certain sections of each flue tapered in opposite directions. This makes it possible for a man to get between the various flues and enables the whole of their external surface to be thoroughly scraped and inspected.

2. The Cornish Boiler.—When a smaller rate of evaporation is required than can be economically maintained with even the smallest Lancashire boiler, the Cornish boiler is frequently used. It consists of a plain cylindrical shell with flat end plates and one furnace tube. In all other respects it resembles the Lancashire boiler and may or may not be fitted with Galloway cross tubes in the furnace tube. The diameter of the shell varies from about 4 feet 6 inches to 6 feet, depending upon the rate of evaporation required, and the length varies from three to four diameters. The usual working pressure is about 80 pounds per square inch, and the furnace tube is usually one-half the diameter of the shell. Fig. 37 shows a cross-section of a Cornish boiler in its brickwork setting for which the general proportions given for the Lancashire boiler apply.

The usual proportions of Cornish boilers named above give about 25 square feet of heating surface per square foot of grate area, and with ordinary chimney draught they will evaporate from 150 pounds of water per square foot of grate area per hour in the smaller sizes to about 170 pounds in the larger sizes. The remarks as to large water capacity, steaming qualities, accessibility for cleaning and repairs already made in connection with Lancashire boilers apply equally well to Cornish boilers.

3. Lancashire and Cornish Multitubular Boilers.—In cases where a high rate of evaporation is required in conjunction with a high efficiency the heating surfaces of ordinary

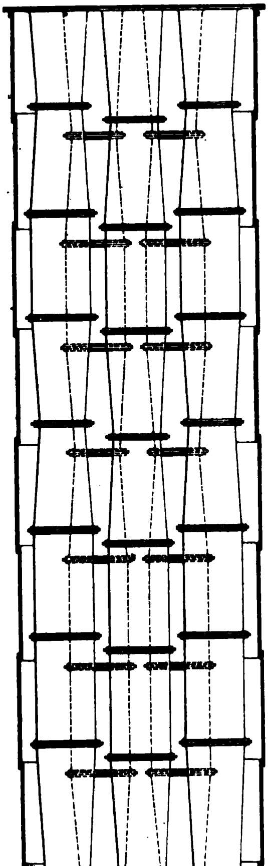


FIG. 40.—Longitudinal section of a five-flued boiler.

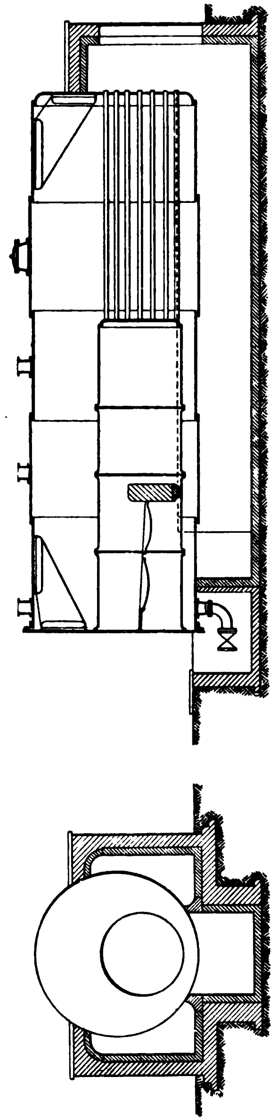


FIG. 41.—Cornish multitubular boiler.

Lancashire and Cornish boilers are occasionally increased by replacing about one-third or one-half of the length of the furnace tubes with a number of small smoke tubes from $2\frac{1}{2}$ to 3 inches in diameter. The cost of manufacture of this type of boiler is higher than that of the ordinary Lancashire or Cornish boiler, and it is not usually constructed for a higher working pressure than 120 pounds per square inch. Fig. 41 shows the construction of the modified Cornish boiler as made by Messrs. Marshall, Sons and Co., Ltd., of Gainsborough.

Boilers of this type are installed in a brickwork setting like that of the ordinary Lancashire and Cornish type, but provision must be made at the back end to facilitate the cleaning of the smoke tubes. The products of combustion pass along the furnace tube, through the smoke tubes down into the flue beneath the boiler, and thence along it to the front end where they divide and pass along the two side flues to the back of the boiler, and finally to the chimney. If scale or incrustation is deposited on the smoke tubes it is difficult to remove, the tubes being numerous and close together, and in addition to the loss of efficiency and the risk of overheating the tubes there is a great tendency to cause leakage at the tube plates. For this reason it is not desirable to use dirty or hard feed-water. For the higher pressures and largest sizes Messrs. Marshall, Sons and Co. put in a corrugated furnace tube as shown in Fig. 42. This boiler is 7 feet diameter and 19 feet long and evaporates about 4000 pounds of water per hour.

Another type of combined Cornish and multitubular boiler is shown in Fig. 43. In this design the set of smoke tubes (the multitubular portion) is placed above a short Cornish (or Lancashire) boiler, thereby considerably increasing the heating surface. On the top of the multitubular portion a steam drum is fixed to which all the usual fittings are attached. This type of boiler is largely used abroad, in countries where the carriage of large pieces is a matter of great difficulty. The lower portion is always kept full of water, the water level being such as to cover all the smoke tubes, and the steam, as generated, ascends into the steam space in the multitubular portion and thence into the steam drum at the top. The ample water capacity in the

lowest section where scale is most likely to accumulate is a

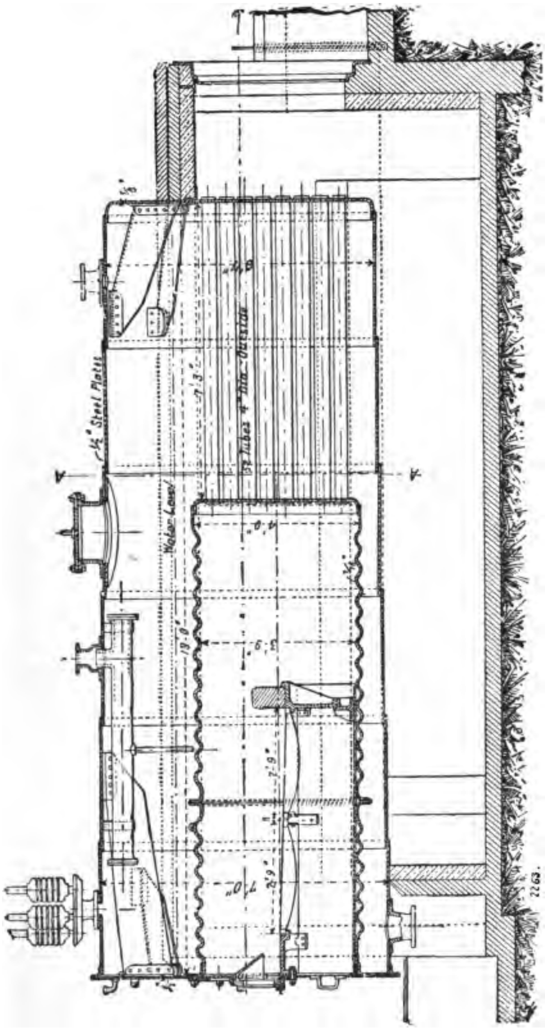


FIG. 42.—Cornish multitubular boiler (Messrs. Marshall, Sons & Co., Ltd.).

good feature in the design as it facilitates cleaning. The path of the gases is first through the furnace tube in the Cornish

portion, then through the smoke tubes above, and thence to the chimney. Doors are fitted in the brickwork setting which when opened expose the ends of all the smoke tubes to view for inspection and cleaning purposes.

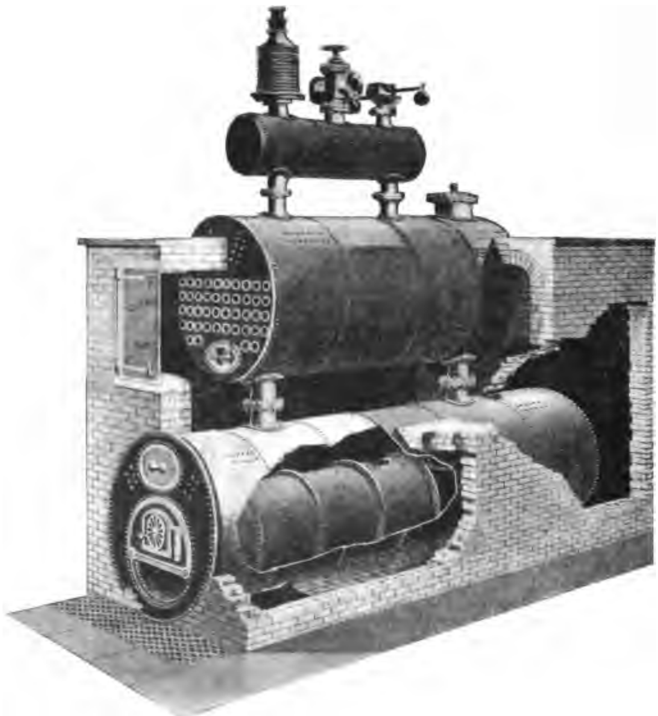


FIG. 43.—Combined Cornish and multitubular boiler.

4. The Yorkshire Boiler.—The construction of this type of boiler will be readily understood by reference to Fig. 44 which shows the shell of a Yorkshire boiler 8 feet 6 inches diameter and 20 feet long. It is much shorter than a Lancashire boiler of the same diameter, and the two internal furnace tubes are tapered, increasing uniformly in diameter, and set with

a slight inclination upwards from front to back. By this means the heating surface per foot length of the tube increases as the temperature of the hot gases decreases, and it is claimed that a more uniform transmission of heat is obtained, which results in a greater evaporation per square foot of heating surface than is obtained in the Lancashire boiler. It may, however, be questioned whether this claim is borne out to any considerable degree in practice. The tubes at the furnace end are smaller in diameter than the tubes of a Lancashire boiler of the same

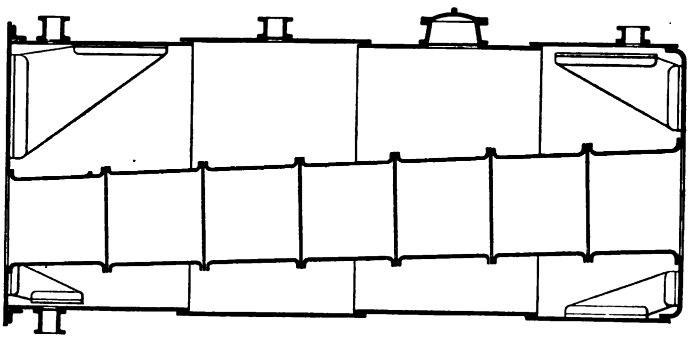


FIG. 44.—Yorkshire boiler.

external diameter, hence the grate area must be smaller also, which results in less coal being efficiently burned in the same time. Practical tests on similar types of boilers show that other things being equal, that boiler is most efficient which has the largest heating surface relatively to the fuel consumed, and that similar boilers worked under similar conditions may naturally be expected to give similar results. The boiler is installed in a brickwork setting exactly similar to that of a Lancashire boiler.

5. The Galloway Boiler.—In this boiler, the two furnace tubes A and B (Fig. 45) unite at a short distance behind the firebridges into one common tube C, which is oval in cross-section and runs through to the back end of the boiler. This oval tube is strengthened by a number of Galloway cone tubes

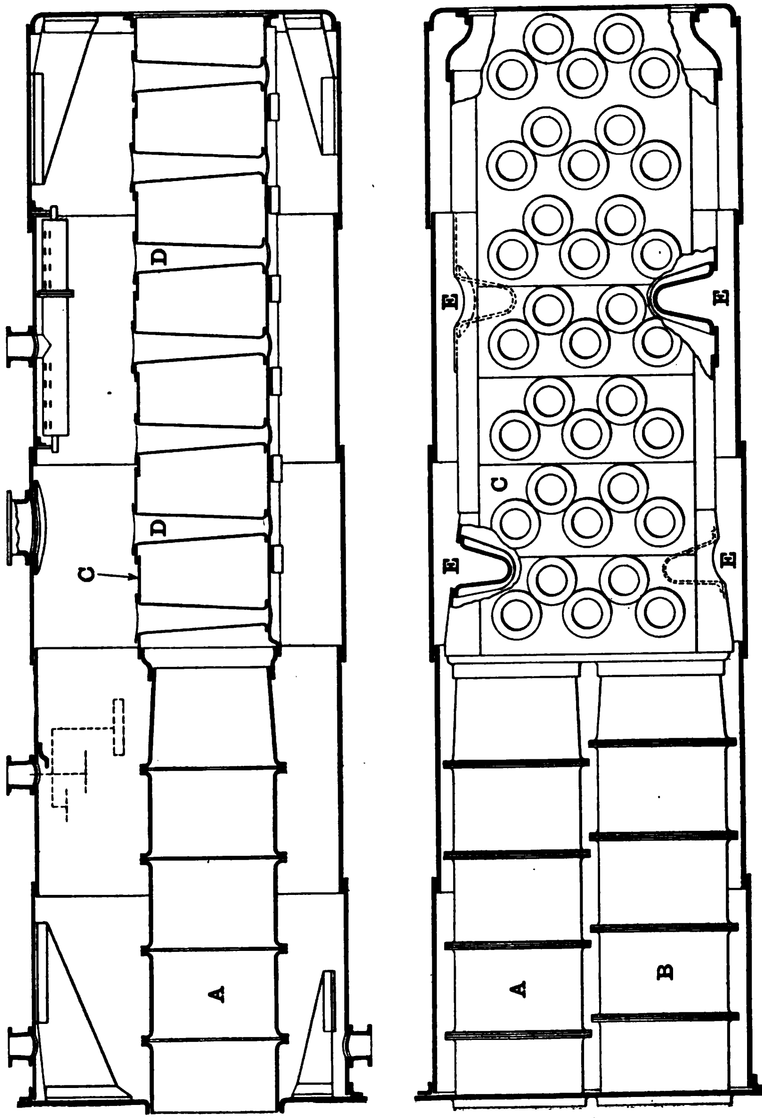


FIG. 45.—Galloway boiler.

D, all of which are tapered and interchangeable. Four pockets E are made in C in order to divert the gases amongst the cone tubes, and together with the cone tubes greatly increase the heating surface of the boiler. The two other pockets at the

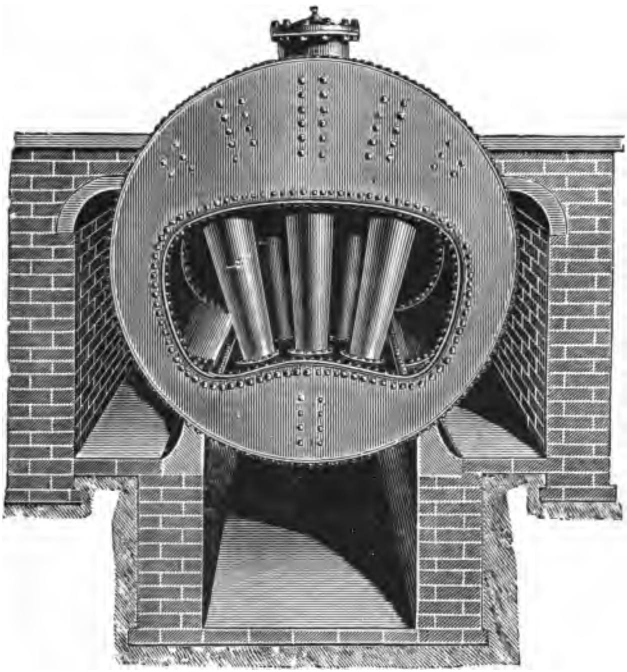


FIG. 46.—Back view of Galloway boiler.

back end of the oval tube, together with the other four, give the necessary amount of flexibility in order to allow for expansion and contraction due to differences of temperature. The boiler illustrated is the latest type of Galloway boiler, and is designed for a working pressure of 140 pounds per square inch. Fig. 46 illustrates the back view of the boiler and shows clearly the cone tubes and brickwork setting.

The boiler is also made to work with external furnaces, in which case the oval flue tube is carried through from one end

of the boiler shell to the other, the furnaces being arranged as already described (Figs. 20 and 22, pp. 65 and 68).

6. Internally Fired Multitubular Boiler.—This type of boiler (Fig. 47) allows a very large heating surface to be arranged in a small space, resembling in this respect both the Scotch marine boiler (Art. 7) and the locomotive (Art. 10). It is much shorter than the Lancashire boiler but may be of larger diameter, and may have one, two, or three furnaces of the same type as those of the Lancashire boiler. The gases from the furnaces pass along the furnace tubes, and are delivered into a combustion chamber which is built either of brickwork or of iron plates and lined with firebrick. In this chamber the unburnt gases can mix freely, and the high temperature at which the firebrick lining is maintained tends to assist the completion of combustion. The gases pass from the combustion chamber through a number of smoke tubes to the smoke box in front of the boiler and thence through a downcast flue to the chimney, or in the case of a marine boiler from the smoke box up the funnel as indicated in Fig. 47.

In order to prevent the front and back plates from bulging, longitudinal stays are fitted as shown, there being no room for the gusset stays used for this purpose in the Lancashire boiler. A certain number of the smoke tubes also act as longitudinal stays and are made thicker than the others; such tubes are screwed on their enlarged ends and fitted with nuts.

This type of boiler is known by various names, viz.: The "Dry Back," "Return Tubular," and "Economic" boiler. If necessary, the heating surface can be further increased by installing the boiler in a brickwork setting similar to that of the Lancashire boiler, in which case the products of combustion pass through the furnace tubes into the combustion chamber, then through the smoke tubes to the smoke box at the front of the boiler, from which they pass along the outside flue between the brickwork setting and the boiler shell on their way to the chimney.

A battery of four Economic boilers installed in a brickwork

setting by Messrs. Davey, Paxman & Co. Ltd., is shown

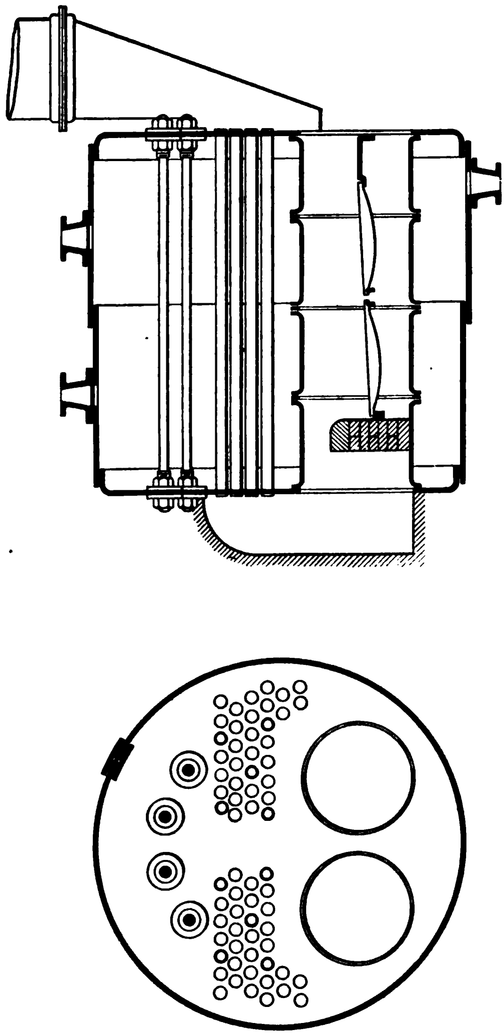


FIG. 47.—Dryback or Economic boiler.

in Fig. 48. The front and side elevation are shown in Fig. 48, the plan view being given in Fig. 48A. The arrangement

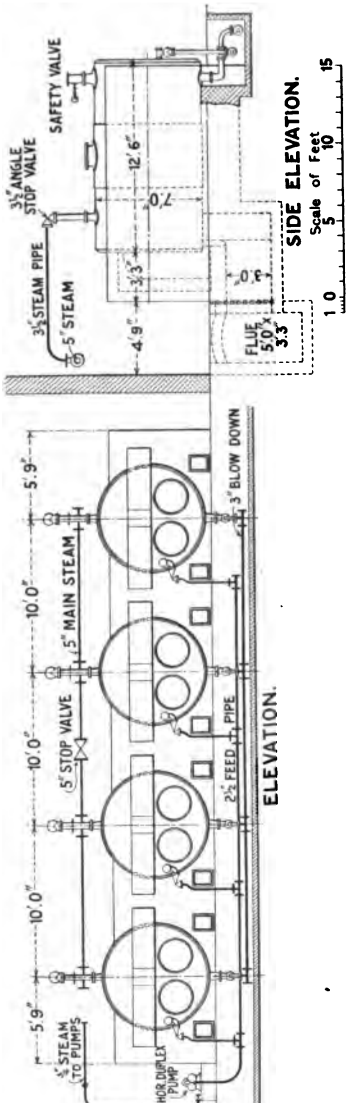


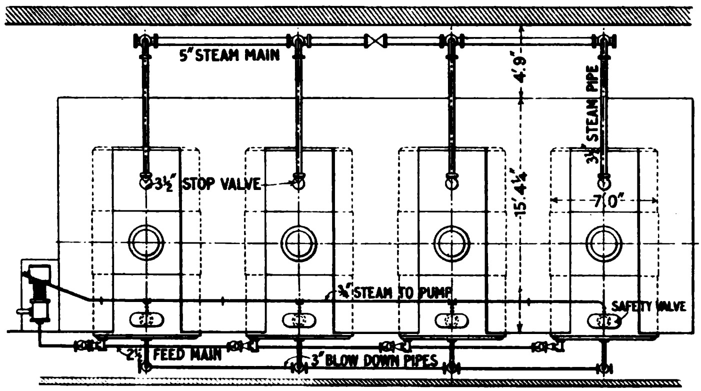
FIG. 48.—Arrangement of four Economic boilers (elevations).

of steam pipes is clearly shown, a particularly good feature consisting in the position of the stop valve of each boiler. It will be noticed from the side elevation that each stop valve is placed at the top of the vertical pipe from the boiler, being thereby situated at the highest point of the piping. By this arrangement it is impossible, when the boiler is not at work, for water to collect in the pipe *above* the stop valve, and therefore there is no danger, through water hammer, when putting any boiler to work on the main steam range. (See Art. 16, Chap. IX.)

The feed pipe is taken from the feed pump along the hearth pit, and by means of the branch pipes and tee pieces shown, the feed water is delivered to the check valve of each boiler; the arrangement of the necessary steam pipe for working the pump is also clearly shown in outline. The blow-off cock of each boiler is connected by means of branch pipes, and tee

pieces to a common blow-down pipe which runs alongside the feed pipe in the hearth pit.

The shell of each boiler is 7 feet 6 inches in diameter and 12 feet 6 inches long, the overall length of the battery (over brickwork) is 41 feet 6 inches, and the width (including hearth pit) is 18 feet 8 inches. Messrs. Davey, Paxman & Co., Ltd., state that with their design of boiler, the flue gases leave at a temperature of 450° F., which enables a very high efficiency to be maintained without the necessity of installing an economiser.

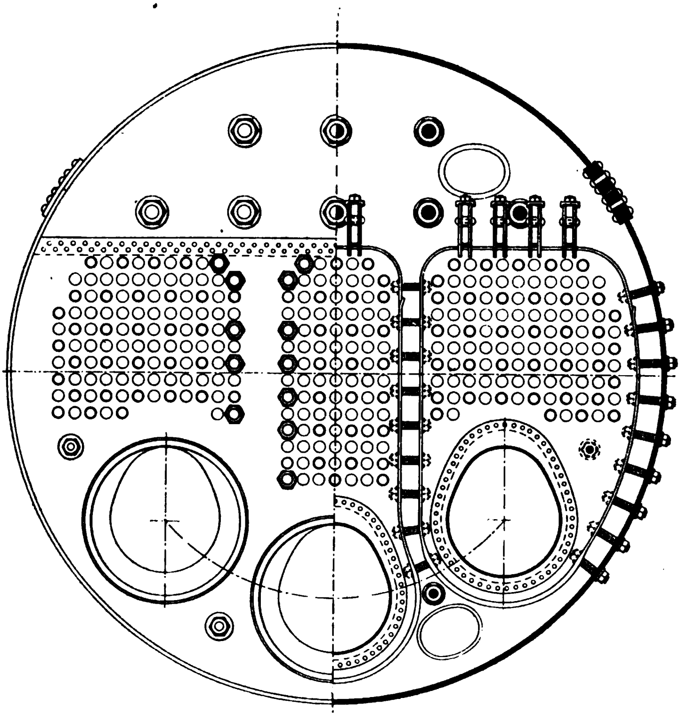


PLAN.

FIG. 48A.—Arrangement of four Economic boilers (plan).

7. Multitubular Marine Boilers.—This type, commonly known as the "Scotch Marine Boiler," is similar to the Dry Back boiler described above, and differs from it chiefly in having an internal combustion chamber or chambers. Figs. 49 and 49A show a single-ended marine boiler of this type as constructed by the North-Eastern Marine Engineering Company, Wallsend. The furnaces are three in number and are made corrugated, each furnace having a separate combustion chamber at the back. The back plate of the boiler and the back of each combustion chamber are securely stayed together by the stay bolts shown, whilst the front of the combustion chamber is stayed to the front plate of the boiler by some of the smoke-tubes

as shown. Eight longitudinal stays, each $3\frac{1}{2}$ inches diameter, connect the front and back plates of the boiler above the combustion chambers, and two longitudinal stays, each $2\frac{1}{2}$ inches diameter, perform the same duty between and below the



Constructed by the North-Eastern Marine Engineering Co. Total heating surface, 2033 sq. ft.; steam space, 350 cub. ft.; working pressure, 180 lbs. per sq. inch. Approx. scale, $\frac{1}{4}$ inch = 1 foot.

FIG. 49.—Scotch marine boiler.

furnaces, whilst the flat top of each combustion chamber is strengthened by means of the roof stays shown. Short-screwed stay bolts also tie the two outer combustion chambers to the boiler shell, and to the middle chamber.

A new departure in this boiler lies in the construction of

the combustion chambers, the bottoms of which have to be made of thicker plates in order to withstand the greater wear

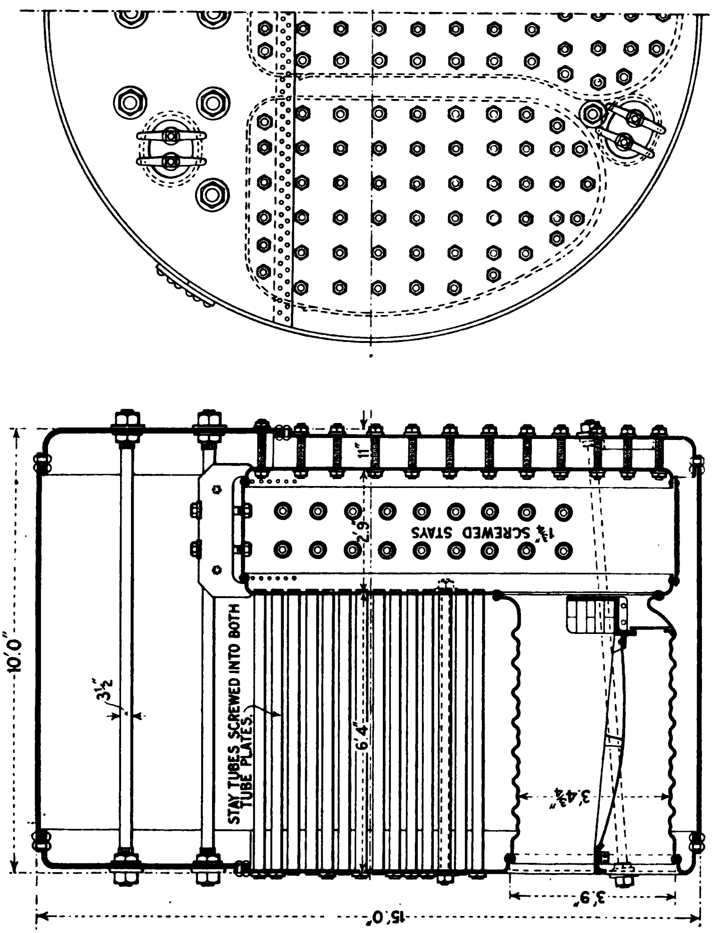
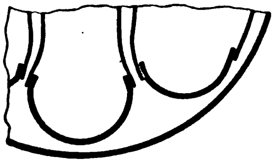


FIG 49A.—Scotch marine boiler.

and tear to which they are subjected when at work. The ordinary method of construction is shown in Fig. 50 in which the side plates of each combustion chamber are made in two

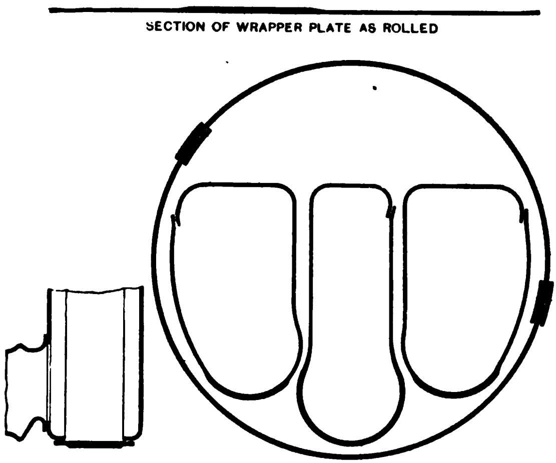
pieces, and riveted to a thick bottom plate for the extra strength required at this part.



Side plate in two pieces, thick bottom plate for extra strength required at this part.

FIG. 50.—Ordinary combustion chamber bottom.

In the new method of construction, only one plate is used for the purpose, and this plate, having in course of manufacture

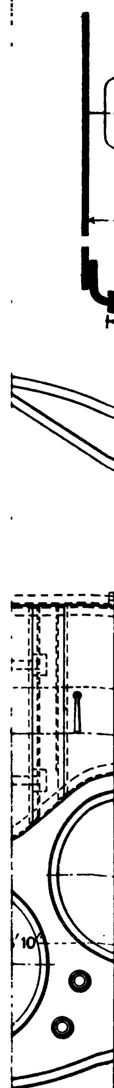


Section of wrapper plate, showing thickened bottom thinned down in way of joints.

Side plate rolled in one piece, with thickened bottom to give extra strength and dispense with joints at this part.

FIG. 51.—Wrapper plate, North-Eastern Marine Engineering Co. New style.

been rolled of varying thickness, is so bent or formed as to have the thicker portion at the bottom, or more exposed portion of the chamber where extra strength is required, while the



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thinner portions from the sides and top as shown in Fig. 51. Only one joint is thus required, and this may be placed near the top of the chamber, where it is free from the disturbing effect of the expansion and contraction of the furnaces, and of the more or less stagnant water at the bottom of the boiler below the furnace level. By this method of construction the usual three-ply joints necessitated by the use of separate thick and thin bottom and side plates are entirely avoided, while the change from one thickness to another being very gradual, no sharp or sudden variations of thickness occur at any part.

A similar type of boiler of approximately the same size by the Central Marine Engine Works, West Hartlepool, is shown in Plate I.

The general proportions of this class of boiler vary considerably, the ratio of length to diameter being altered to suit the space available. The diameter ranges from about 6 feet in small sizes with one furnace, to as much as 18 feet in the largest sizes with three or four furnaces. The length is usually not more than one and a quarter times the diameter even in the small sizes, and in the larger sizes it is frequently less than the diameter; *e.g.* in Fig. 49 the diameter is 15 feet and the length 10 feet, and in Plate I the diameter and length are 15 feet and 10 feet 6 inches respectively.

In the mercantile marine where the load on the boilers is more or less uniform this type of boiler is almost universally used, and has proved itself an economical and reliable steam-generator. An efficiency as high as 82 per cent. has been obtained with the single-ended boilers of the Cunard liner *Saxonia*.* These boilers were fitted with Howden's system of forced draught, and the above efficiency was obtained with a coal consumption of 20·6 pounds per square foot of grate per hour, and an evaporation of 12·3 pounds of water per pound of coal from and at 212° F., the average temperature of the funnel gases being only 396° F., while 23·4 pounds of air were supplied per pound of coal.

As in the case of the Lancashire boiler it is not advisable to

* See *Report of Admiralty Committee on Water-tube Boilers*, Vol. I., reprinted in *Engineering*, 28th Feb., 1902, p. 278.

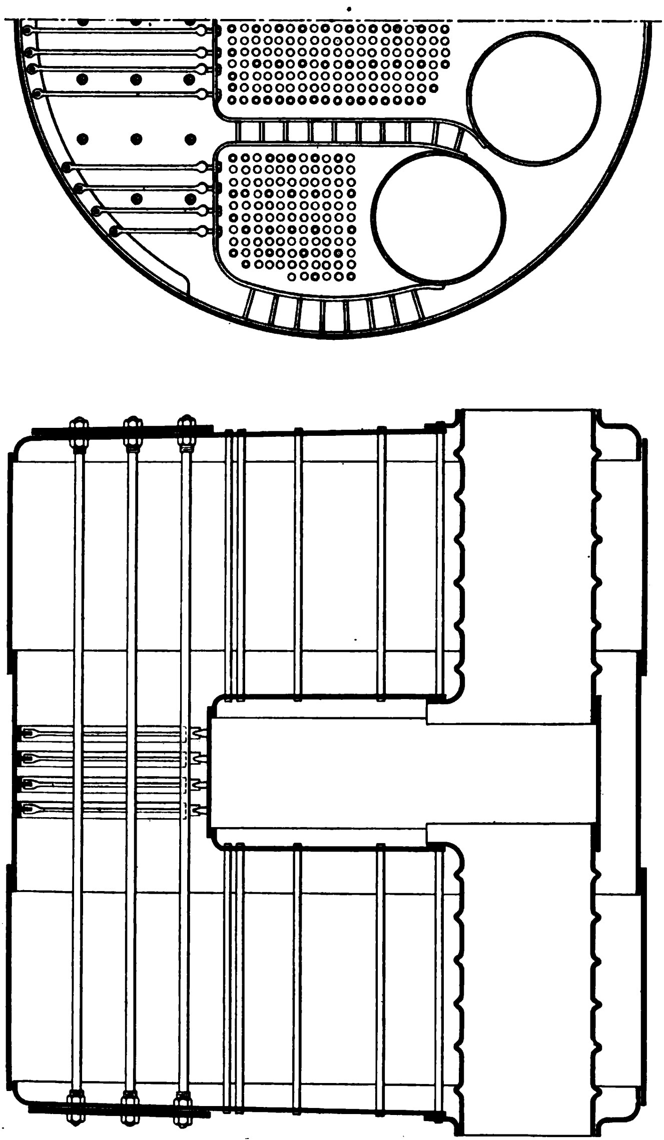


FIG. 52.—Double-ended marine boiler.

raise steam very quickly in the Scotch marine boiler, for fear of straining it.

8. Double-Ended Marine Boiler.—The usual type of multitubular marine boiler is the single-ended one described above; occasionally, however, the large sizes for greater evaporation are made double-ended as shown in Fig. 52. By this design weight and cost is saved as compared with a single-ended boiler of the same power.

9. Externally Fired Multitubular Boiler.—One form of this type of boiler as made by Messrs. Marshall, Sons & Co., Ltd., is shown in Fig. 53. It consists of a cylindrical shell with flat ends containing a number of smoke tubes in the lower portion passing from end to end, and communicating at the front end of the boiler with a smoke box and chimney. The gases from the furnace pass along the bottom of the boiler, and return through the smoke tubes to the chimney.

This type of boiler is cheap and compact and is largely used in countries where transport is a serious item. It is a good and efficient steam generator, but has the disadvantage of *all* externally fired boilers that sediment is very liable to accumulate on the bottom plates of the boiler causing overheating and burnt plates. For this reason and also on account of the smoke tubes it should only be employed when the feed water is pure and deposits little sediment, or when a water-softening plant is used in conjunction with hard water.

10. Locomotive Boilers.—There is a great variety of designs of locomotive boilers which differ more or less in details only. Fig. 54 illustrates the boilers used on the Compound Express Passenger Engines of the Midland Railway.

The boiler illustrated is constructed for a working pressure of 220 pounds per square inch. The outside cylindrical shell or barrel is made of steel plates $\frac{5}{8}$ inch thick, the average diameter being 4 feet 8 inches and length 11 feet 11 inches. The circumferential seams are double-riveted lap joints, and the longitudinal seams treble-riveted butt joints with two cover plates. All rivet holes are drilled $\frac{15}{16}$ inch in diameter, the diameter of the rivets also being $\frac{15}{16}$ inch. The smoke box tube

plate is of steel $\frac{13}{16}$ inch thick, the firebox tube plate is $\frac{7}{8}$ inch thick, and the distance between the two tube plates is 12 feet 3 inches. The firebox is of the "Belpaire" type being made of

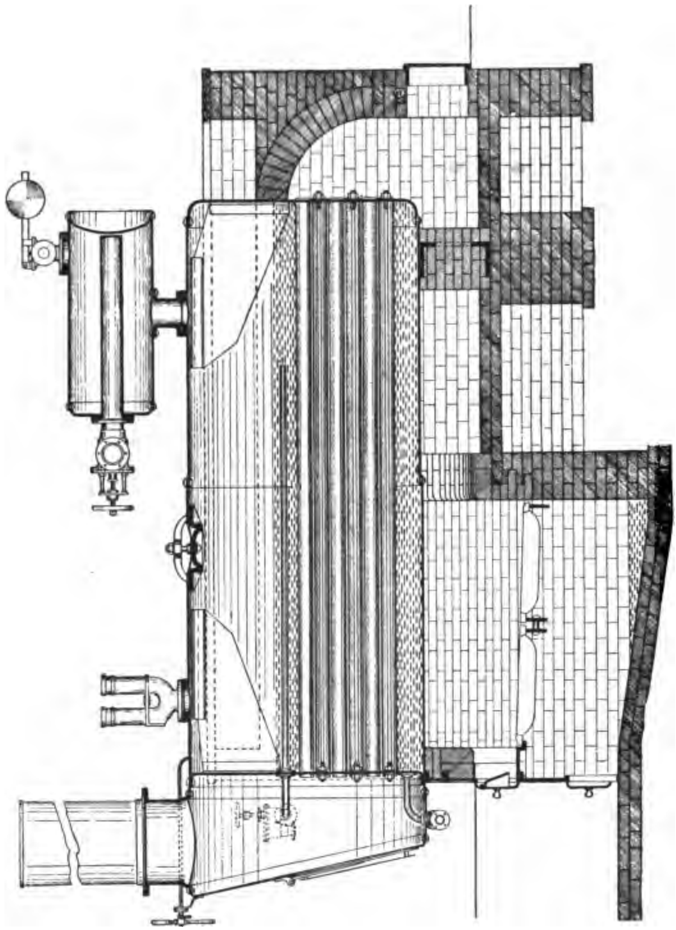


FIG. 53.—Externally fired multitubular boiler.

steel plates $\frac{9}{16}$ inch thick. The flat plates of the firebox are stayed to the outer casing as follows: Screwed stay bolts 1 inch diameter, pitch 3·35 inches, tie the sides of the firebox to the

outer casing, the water space between the firebox and casing being 3 inches. The roof of the firebox is tied to the outer shell by means of about 220 screwed stays $1\frac{1}{4}$ inch diameter as shown, this method being more convenient for boilers of large size than the ordinary roof stays holding the tops of the combustion chambers of the marine boiler shown in Fig. 49.

Tying the back plate of the boiler to the smoke-box tube plate, above the boiler tubes, are 10 longitudinal stays each $1\frac{1}{2}$ inches diameter, and above the firebox are 24 cross stays, each $1\frac{1}{2}$ inches diameter, which stay the sides of the outer casing at this place.

The tubes, 216 in number, are $1\frac{7}{8}$ inches external diameter, but are enlarged at the smoke box end for a length of about 3 inches. The total heating surface is 1456 square feet, 1305·5 square feet of which are in the tubes, the remainder being furnished by the firebox.

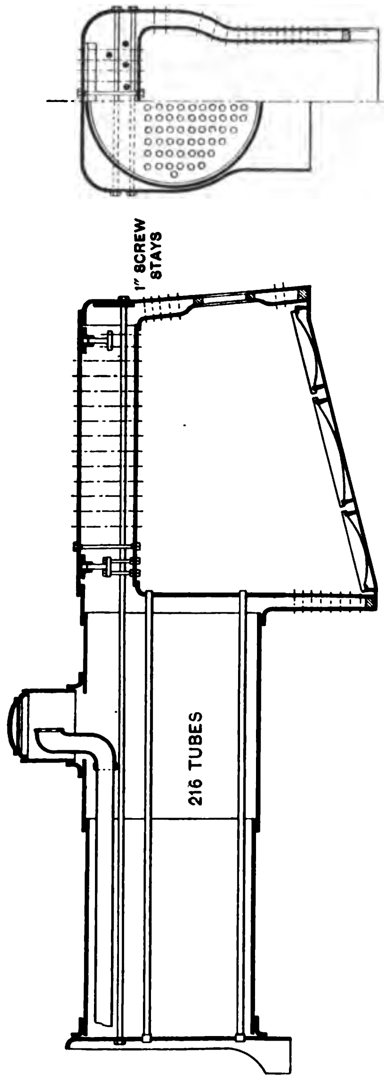


FIG. 54.—Locomotive boiler (Midland Railway). Approx. scale, $\frac{1}{3}''$ = 1 ft.

The length of the boiler (excluding the smoke-box) from the smoke-box tube plate to the rear of the firebox is 21 feet.

The firebars are supplied in three lengths the front and rear sections being each 2 feet 4 inches long, whilst the middle section is 3 feet 7 inches in length. The width of the grate is 3 feet 4 inches giving a grate area of 28·4 square feet. The ratio of heating surface to grate area is thus 51·2.

A steam dome, 1 foot 11 inches diameter and 1 foot 9 inches high, is fitted from which the steam is taken through the regulator and a 6-inch pipe to the engine cylinder by way of the smoke-box.

Locomotive Boiler Tubes.—There is a great similarity in the general practice with regard to boiler tubes, although small differences will be frequently met with on different railways, owing often to variations in the fuel and water used by the boilers, and in the work upon which the engines are engaged. The usual sizes for tubes are about $1\frac{3}{4}$ inch outside diameter for the smaller boilers, and 2 inches or even larger for the large boilers now common on most railways. The distance at which the tubes are pitched from each other needs careful attention, since, if too close, the difficulty in dislodging scale will be increased and the circulation of the water impaired. Also the risk of cracking the tube plates is greater when there is very little metal to resist the expanding of the tubes and the variations of temperature as the boiler is alternately heated and cooled in the conditions of service. If the tube plates are too rigid, either because of the tubes being crowded or by the whole tube area being too close to the outer angles and flanges of the smoke-box tube plate, there will be difficulty experienced sometimes, as the tubes will either have to force the plate out or move in the tube holes, and the risk of leakage is very great.

The best practice is to allow as much space as possible in the front tube plate outside the tube area, so that this plate can give with the tubes as they lengthen and shorten with their variations of temperature; the tubes are thus not compelled either to move a too rigid plate out and in, or to slide in the holes which is the only alternative if the plates will not give with them.

The materials employed for boiler tubes are copper, brass, iron

and steel. The first two are still largely used, but there appears a tendency for the latter two to replace them on many railways. The quality of the metal, whichever is chosen, is of great importance; only the best of its kind for the purpose will give satisfactory results. If inferior brands are used, leakage becomes a matter of frequent occurrence, because they will not stand the severe conditions of fixing them in the tube plates or the work which they have to do when they are fixed.

The tubes are made parallel in the central part, but enlarged a little at the smoke-box end for about 3 inches. This is done so that the holes in the smoke-box tube plate may be made larger and so allow the tubes to be easily put in place, but more particularly so that they may be taken out when coated with scale. With brass and copper tubes the firebox end is made of the same size as the body of the tube, but if of iron or steel it is often the custom to reduce this end a little in diameter; no ferrules are used, and by this reduction in size the same flue area is allowed as if ferrules were used, while the amount of tube plate between the tubes is increased by the same amount and the strength of the plate with it.

The holes in the tube plates are made parallel and it is not customary, as a rule, to put stays in the tube area, although some makers do so. In some cases also a slight taper in the tube-holes is made, but this is not a general custom, as it has been found that a well fixed tube in a parallel hole acts effectually as a stay for this part of the boiler.

The tubes are put in from the smoke-box end, and at that end are simply expanded into their respective holes in the smoke-box tube plate. The firebox end of each tube in addition to being expanded is beaded over as shown at A (Fig. 55), and if of the softer metals, copper or brass, steel ferrules B are also put in to help to support the tubes in the holes. It is not found necessary to use ferrules with iron or steel tubes. The beading is done in order to remove the sharp ends of the tubes which would afford lodgment for soot and that peculiar formation known as "bird's nests." The main reason, however, for beading is to protect the ends of the tubes from being burned off by the fire as well as to assist in making a tight joint.

American Locomotive Boiler.—The large gauges on American railways permit the use of larger boilers than are possible in this country. Fig. 56 illustrates the type of boiler used on the New York Central and Hudson River Railroad.

The boiler is constructed of steel, the average diameter of the barrel being 6 feet 10 inches, the plates being $\frac{13}{16}$ inch thick. The smoke-box tube plate is $\frac{5}{8}$ inch thick, the firebox tube plate is $\frac{1}{2}$ inch thick and the distance between the tube plates is 15 feet. The firebox is constructed of steel plates $\frac{3}{8}$ inch thick; screwed stay bolts 1 inch diameter, pitch 4 inches, tie the sides of the firebox to the outer casing; the roof of the firebox is tied to the outer

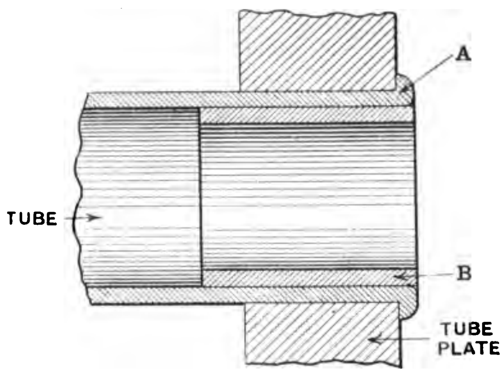


Fig. 55.—Ferrule in firebox end of tubes.

shell by stays 1 inch diameter similar to the boiler above described.

The boiler tubes, 444 in number, are 2 inches external diameter; the total heating surface is 3670 square feet, of which 3460 square feet are in the tubes, the remainder being furnished by the firebox. The firebrick arch is carried on four 3-inch water tubes which connect the front and back of the firebox as shown. The length of the boiler (excluding the smokebox) from the smoke-box tube plate to the rear of the firebox is 25 feet 3 inches. The length of the firegrate is 9 feet 8 inches and width 5 feet 11 inches, giving a grate area of 57 square feet. The ratio of heating surface to grate area is thus 64·3. A new type

of locomotive boiler recently put into operation on the Southern Pacific Railway, U.S.A., and made by the Baldwin Locomotive Works, of Philadelphia, is of interest.* The boiler is made in two parts, the barrel being built in two distinct sections of which the front section is detachable from the back one. The front portion is used as a feed water heater, the feed entering at the bottom and leaving at the top, whence it is taken by external pipes on either side to the boiler through clack valves placed on a horizontal diameter of the boiler barrel. A clack valve is also placed at the heater end of these connecting pipes. The boiler is arranged for firing oil fuel and under normal conditions the engine runs cab first, thus permitting an unobstructed view for the driver. The leading dimensions of the boiler are as follows:—The

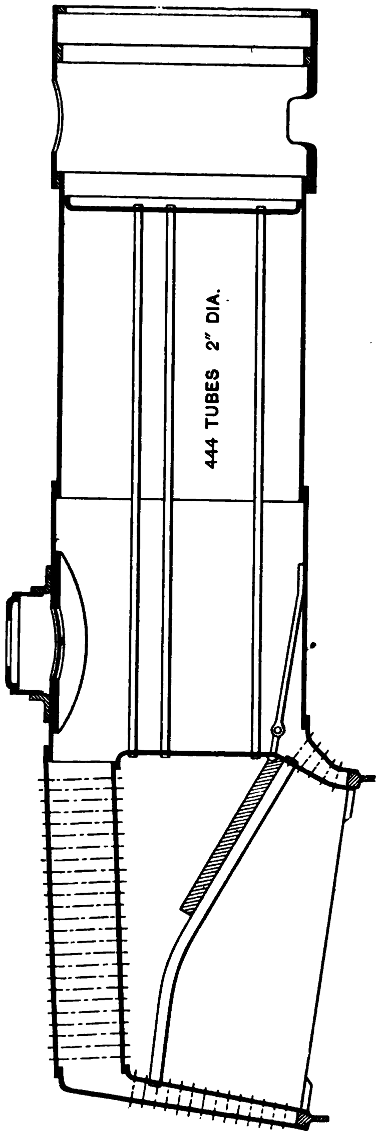


FIG. 56.—Locomotive boiler, American type. Approx. scale, $\frac{1}{5}''$ = 1 ft.

* See *Engineering*, Dec. 15, 1911.

boiler barrel is constructed of steel plates $\frac{3}{4}$ inch thick, the mean diameter being 6 feet 10 inches. The length of the steel firebox is 10 feet $0\frac{5}{8}$ inch, width 7 feet, thickness of tube plate $\frac{1}{2}$ inch, other plates $\frac{3}{8}$ inch. The minimum width of the water space between the firebox and barrel is 5 inches. In the boiler proper there are 495 tubes each 20 feet 6 inches long and 2 inches external diameter; in the front section, or feed heater portion, there are 424 tubes each 6 feet 3 inches long and $2\frac{1}{4}$ inches external diameter. The heating surface is, in the firebox 235 square feet, in the boiler tubes 5292 square feet, and in the feed heater tubes 1590 square feet, giving a total of 7117 square feet. The grate area is 70 square feet, giving a ratio of total heating surface to grate area of 101·7, or for the boiler portion only (omitting the heating surface of the feed heater tubes) a ratio of 75·6. The working pressure is 200 pounds per square inch. The total weight of engine and boiler is 171·8 tons, of which 142·9 tons is on the driving wheels. The weight of engine and tender together when in running order is 253·57 tons.

11. Vertical Boilers.—Vertical boilers are usually constructed in small sizes only, being commonly employed in connection with steam cranes and other portable steam machinery. On account of the small floor space required and the absence of brickwork setting they are readily applicable to this class of work, but they are not very economical steam generators owing to the comparatively small ratio of heating surface to grate area. Fig. 57 shows the construction of the Cochran Multitubular Boiler, which is selected as a good example of this class of boiler. The boiler illustrated is 6 feet 6 inches in diameter and 14 feet high, with a heating surface of 450 square feet and a grate area of 22·50 square feet, giving a ratio of heating surface to grate area of 20. The boiler is fitted with 143 smoke tubes each $2\frac{1}{2}$ inches external diameter, the funnel being 2 feet diameter and 30 feet high. When working on full load under favourable conditions the boiler shown will burn 23·7 pounds of coal per square foot of grate area and evaporate about 3100 pounds of water per hour at a pressure of 100 pounds per square inch, corresponding to an evaporation of 7 pounds of water per hour per square foot of heating surface.

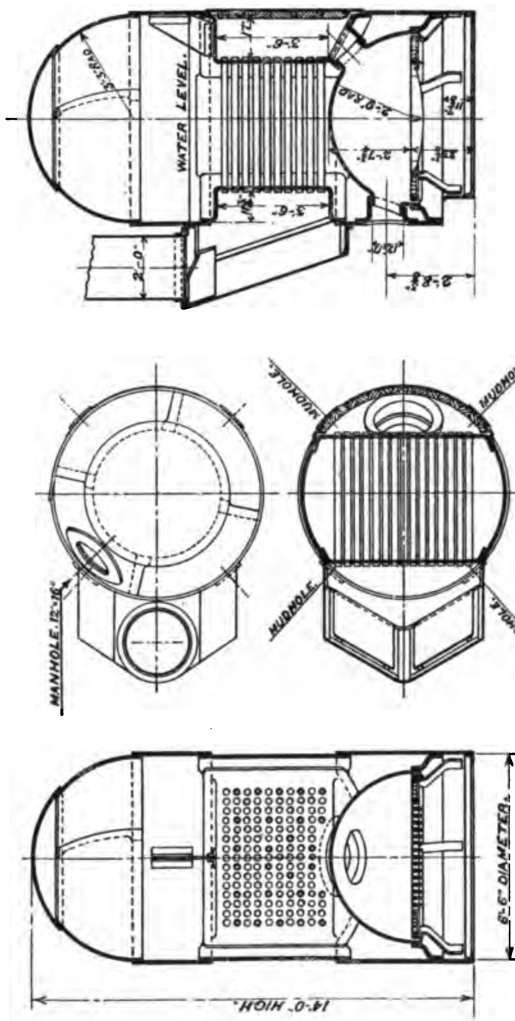


FIG. 57.—Cochran vertical multitubular boiler.

CHAPTER VII

WATER-TUBE BOILERS

1. General Considerations.—In order to obtain the maximum of power, combined with economy, from the minimum weight and dimensions of a steam engine, it is necessary in modern practice to use high-pressure steam. Modern practice also demands in many cases the ability to raise steam rapidly, particularly for naval purposes and in electric generating stations. This demand has resulted in the "water-tube" boiler, but the question as to whether their advantages outweigh those of the cylindrical or smoke-tube type depends entirely on the requirements of the particular service for which they are intended. The essential difference between the "water-tube" or "tubulous" type of boiler and the smoke-tube type previously considered, is that the steam and water are contained *inside* a system of tubes of comparatively small diameter, the fire and hot gases being outside these tubes; also, the outside casing of the boiler is *not* subject to pressure, as is the case with the shell of the Lancashire or Scotch marine boiler and their modifications. On account of the tubes being subjected to the internal steam pressure, it is obvious that the metal of which they are constructed is subjected to *tensile* stresses and not to compression, as is the case in the smoke tubes of the Scotch marine boiler and the locomotive boiler, or the furnace tubes of the Lancashire boiler. Also in the water-tube boiler the furnace is invariably external to the boiler proper, although, of course, it is within the boiler casing.

Before proceeding with the description of some of the many different types of water-tube boiler, it will be as well to

mention their advantages and disadvantages, as compared with the cylindrical or smoke-tube type.

2. Advantages of Water-Tube Boilers.—(1) *Higher steam pressures can be employed without danger.* In the larger sizes of the cylindrical boiler, *i.e.* the Lancashire and Scotch marine boiler, the thickness of the shell plates required becomes excessive for even moderately high steam pressures. It should also be remembered that the furnace tubes are subjected to external pressure, and, therefore, the material of which they are constructed carries a compressive stress. Although a cylindrical (or spherical) vessel is of the strongest shape to withstand either external or internal pressure, there is a great difference required between the construction of the shell plates and the furnace tubes. If the outer shell is not truly cylindrical it does not matter much, because being subjected to tension on account of the internal steam pressure, the only effect is to make the shell assume a more truly circular shape in cross section. If, however, the furnace tubes are not truly circular in cross section, the external steam pressure will increase this defect, and such tubes, although strong enough to withstand the same pressure internally, would be unsatisfactory when carrying an external pressure (see also Art. 14, Chap. I.). In the water-tube boiler, however, this difficulty does not occur, and small tubes are used for the heating surface exposed to the fire, these tubes being connected to steam and water drums of comparatively small diameter, which require thinner plates than the large shells of the internally-fired boiler. An example will make this clear.

For a working pressure of 220 pounds per square inch the thickness of mild steel plates required for the steam-drum of a water-tube boiler 4 feet diameter would be about $\frac{11}{16}$ inch, whilst for a marine boiler shell 15 feet diameter, to carry the same steam pressure, the thickness of plates required would be about $2\frac{5}{8}$ inches, allowing a tensile stress in each case of 10,000 pounds per square inch (Art. 14, Chap. I.). Such a thickness and weight of plate is not to be obtained, hence special steel would have to be used, in order to keep down the thickness of the plates to a practical figure (as, for instance, in the case of the boilers of the *Mauretania*).

Again, on account of the small volume of water and steam contained in a water-tube boiler, an explosion, particularly a burst tube, the liberated steam from which can escape up the chimney or funnel, is likely to be attended with less serious consequences than with the Lancashire or marine type of boiler.

(2) *Rapid Steam Raising.*—The water capacity of the boiler is very much less than that of the smoke-tube type; in conjunction with the more rapid circulation of the water and the great freedom for expansion and contraction due to changes of temperature rendered possible by the construction of the boiler, this results in the ability to raise steam very rapidly in cases of emergency, and in a smaller dead weight of boiler for the same rate of evaporation. Also, the boiler can be cooled rapidly in order to carry out small repairs or cleaning without causing the tube joints to leak.

(3) *Convenience in repairs, erection and transport.*—If any part of the boiler should fail, or be damaged in any way, *i.e.* if a tube should burst, it may be replaced in a very short time with little trouble. When used on board ship the various parts can be removed in pieces without opening the deck. Also, the various parts being comparatively small and of light weight, they are easily transported and erected in places where it would be difficult to send a whole cylindrical boiler.

(4) *Less weight and space with larger grate area.*—The water-tube boiler, consisting as it does of elements of small diameter and thickness of metal, is of less weight and occupies a smaller floor space than a cylindrical boiler of the same power. On account of the external furnace, the ratio of heating surface to grate area can be varied within very wide limits, which makes it possible to burn poor fuel (Art. 6, Chap. III.). The external furnace also permits of a larger grate area and combustion chamber, giving every facility for complete combustion before the gases come in contact with the comparatively cool water tubes (Art. 1, Chap. III.), and it conveniently allows for the adoption of mechanical stokers.

(5) Lastly, for land work it may be added that water-tube boilers are built for much larger heating surfaces and evaporative

capacities than Lancashire boilers. By reference to Table I. (p. 7), it will be seen that standard sizes of water-tube boilers are procurable up to 6000–8000 square feet of heating surface and 24,000–30,000 pounds of steam per hour from and at 212° F., as contrasted with a 30 feet × 9 feet Lancashire boiler having a heating surface of about 1170 square feet, and evaporating when forced 12,000 pounds of water per hour.

3. Disadvantages of Water-Tube Boilers.—(1) *Small water capacity.* Although the small water capacity is an advantage from the point of view of rapid steam raising, it is considered by many to be a disadvantage on account of the difficulty sometimes experienced in maintaining the water level uniform. In round figures it may be stated that in the case of a water-tube boiler working at its normal rate, a total interruption in the feed supply would result in the boiler being emptied in about five minutes; in any case, a short interruption will cause a large reduction in the water level. In land practice this may not be altogether a disadvantage, because with a single boiler supplied by a feed pump of the requisite capacity, the frequent sounding of the low-water alarm shows that the stoker is not on the *qui vive*, and the remedy is obvious.

In installations where several boilers are fed from a common feed pipe this is undoubtedly a disadvantage, and in such cases the maintenance of the water level and the control of the feed will need considerable care, unless automatic feed regulators are fitted to each boiler.

(2) *Pure Feed Water necessary.*—On account of the small water capacity and small diameter of the thin-walled water-tubes, it is essential that the feed water should be pure. The tubes are enabled to withstand the high temperature to which they are subjected solely on account of the rapid circulation of water through them. If, therefore, a deposit once forms inside a tube, the circulation is restricted; moreover, when once formed, the deposit tends to increase in bulk, resulting in a burnt tube and failure. For this reason alone, as distinct from its corrosive action, the use of sea water in marine work is not permissible. In this connection it may be well to point out

that if a water-tube boiler be designed so that the water circulation is rapid enough,* it will be impossible for any deposit to form inside the tubes. Sir J. I. Thornycroft found that the circulation in the "Daring" type of boiler (Art. 17) actually polished the insides of the tubes and practically prevented the formation of any deposit.

(3) *Greater danger from corrosion.*—The causes of internal corrosion are fully dealt with in Chapter X. Put very briefly, if air and carbon dioxide are present in the feed water, or if moist air has access to the interior of a boiler, corrosion is almost certain to take place. In condensing plants where the condensed steam from the engines is used as feed water for the boiler, it is important that all greasy matter should be removed. The acids produced by the decomposition of animal or vegetable oils used in lubricating the cylinders are very active in producing corrosion, and for this reason mineral oils (which are non-corrosive) should be exclusively used for this purpose; nevertheless, it is important to remove even mineral oil from the feed water, because, although it does not give rise to corrosion, it forms a brown deposit on the inside of the tubes and plates; this being a very bad conductor of heat results in a lower efficiency, and is likely to cause overheating of the metal plates.†

Protection against corrosion in water-tube boilers is much more important than in the case of the smoke-tube or Lancashire boiler, because the tubes and steam drums being of smaller diameter are thin, and therefore afford a much smaller margin against wasting; this is particularly the case in water-tube boilers of the small-tube type in which the tubes are only about $\frac{1}{16}$ inch thick.

External wasting of the tubes may occur through an accumulation of soot on the cooler parts of the tubes, such accumulations being liable to absorb moisture and so set up corrosion. Protection against this external wasting is afforded

* See Prof. J. T. Nicolson on "Boiler Economics and the Use of High Gas Speeds," *Institution of Engineers and Shipbuilders of Scotland*, 1911.

† See Sir John Durston's Experiments, "Transactions of the Institution of Naval Architects," vol. xxxiv.

by regularly removing any such accumulation from the tubes by means of a jet of compressed air or steam.

In the case of the small-tube boiler there is great diversity of opinion as to the need for galvanising the tubes for protection against corrosion. Sir J. I. Thornycroft condemns galvanising as being unnecessary and injurious since it involves a preliminary pickling, which forms a starting point for corrosion, especially inside the tubes. He recommends instead, the use of zinc slabs inside the boiler in good metallic contact with the plates. Mr. Yarrow on the other hand recommends galvanising, affirming that it increases the durability of the boiler.

4. Classification of Water-Tube Boilers.—Various methods of classifying water-tube boilers have been employed but perhaps the most general method is to divide them into two main classes, namely:—

(1) Those possessing comparatively large tubes 3 inches or more in diameter, this class being known as large-tube boilers, and

(2) Those having small tubes 2 inches or less in diameter, this class being known as small-tube boilers.

The different types of the large-tube boiler which have proved successful in practice are comparatively few in number; a very great number of small-tube boilers have been proposed and tried, but in this book it is only intended to describe a few of the leading types.

Large Tube Boilers.

5. Babcock and Wilcox Boiler, Land Type.—This boiler, constructed entirely of wrought steel, is composed of tubes about 4″ diam. placed in an inclined position sloping downwards from front to back at an angle of 15° to the horizontal and connected with each other and with the horizontal steam and water drum by the end connecting tubes (Fig. 58). The tubes are divided into sections, each vertical row of inclined tubes being expanded at each end into a wrought-steel sinuous header. Each header is forged in one piece (Fig. 59) in such a form that the tubes are "staggered," or so placed that each

horizontal row comes over the spaces in the previous row. The sections thus formed are connected separately to the top steam and water drum and to the mud drum below the back headers, by short almost vertical tubes expanded into bored holes. In the headers opposite each water tube is arranged a hand hole and cover to allow of the tubes being expanded in position and afterwards cleaned and inspected.

The steam and water drum is made of mild steel and is

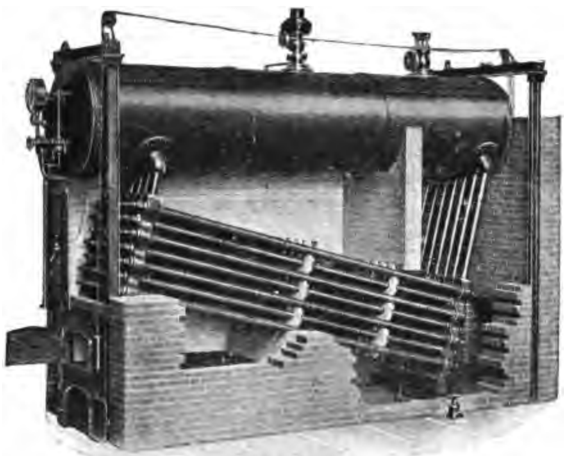


FIG. 58.—Babcock & Wilcox boiler (Land type).

double riveted in the longitudinal seams; it can be made for any desired working steam pressure.

In erecting this boiler it is suspended from horizontal wrought-iron girders resting on upright iron columns as shown in Fig. 58. It is therefore entirely independent of the brickwork setting, and there is no tendency to produce straining of the boiler due to unequal expansion between it and its enclosing brick walls. In addition, it is possible to repair or remove the brickwork when necessary without disturbing the boiler in any way. Two baffling diaphragms of fireclay tiles are fixed between the inclined water tubes as shown, in order to compel the

furnace gases to pass through the spaces between the tubes at right angles.

The boiler shown in Fig. 58 is arranged for hand firing and delivers saturated steam. It is made in various sizes for evaporations of from 360 to 30,000 pounds of water per hour.

The firegrate is under the front and higher ends of the tubes and the products of combustion pass up between the tubes into the combustion chamber under the steam and water drum; from thence they pass down between the tubes (between the two baffling diaphragms of fireclay tiles), thence once more up through the spaces between the tubes and off to the chimney. The water inside the tubes, as it is heated, tends to rise towards the higher front end, and as it is converted into steam—the mingled column of steam and water being of less specific gravity than the solid water at the back end of the boiler—rises through the nearly vertical passages from the front headers into the steam and water drum above the tubes, where the steam separates from the water, the latter flowing back to the rear and down again to the back headers and through the tubes in a continuous circulation. As the passages are all large and free this circulation is very rapid; the steam is swept away almost as fast as it is formed, and the water throughout the boiler is thoroughly intermingled so that the tendency to form deposits on the inside of the tubes is reduced to a minimum and any suspended matter is deposited in the mud drum, whence it can be blown out through the blow-off cock. The steam is drawn off through the stop valve at the top of the steam drum at the back end of the boiler.

Fig. 59.—Forged header for Babcock & Wilcox boiler.

When desired, an internal superheater may be fitted between

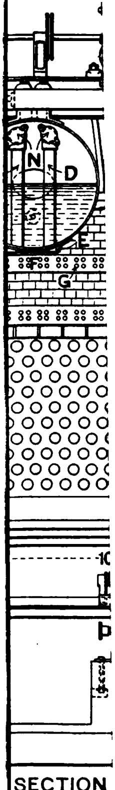
the inclined tubes and the steam drum. When using cheap bituminous coal, and smoke prevention is important, the boiler is usually fitted with a chain-grate stoker. Plate II. shows the latest type of the Babcock and Wilcox boiler fitted with a chain-grate stoker and integral superheater.

6. Babcock and Wilcox Boiler, Marine Type.—This boiler, which is a modification of the land type, is constructed wholly of wrought steel and consists of an arrangement of

FIG. 60.—Babcock & Wilcox marine boiler.

inclined tubes forming the bulk of the heating surface, sinuous headers to which the tubes are attached, a horizontal steam and water drum, a mud drum, and a furnace of large capacity beneath the inclined tubes as shown in Figs. 60 and 61. As in the land type, the inclined tubes are divided into vertical sections, and, to ensure a continuous circulation, are placed at an inclination of 15° to the horizontal.

Extending across the front of the boiler and connected to



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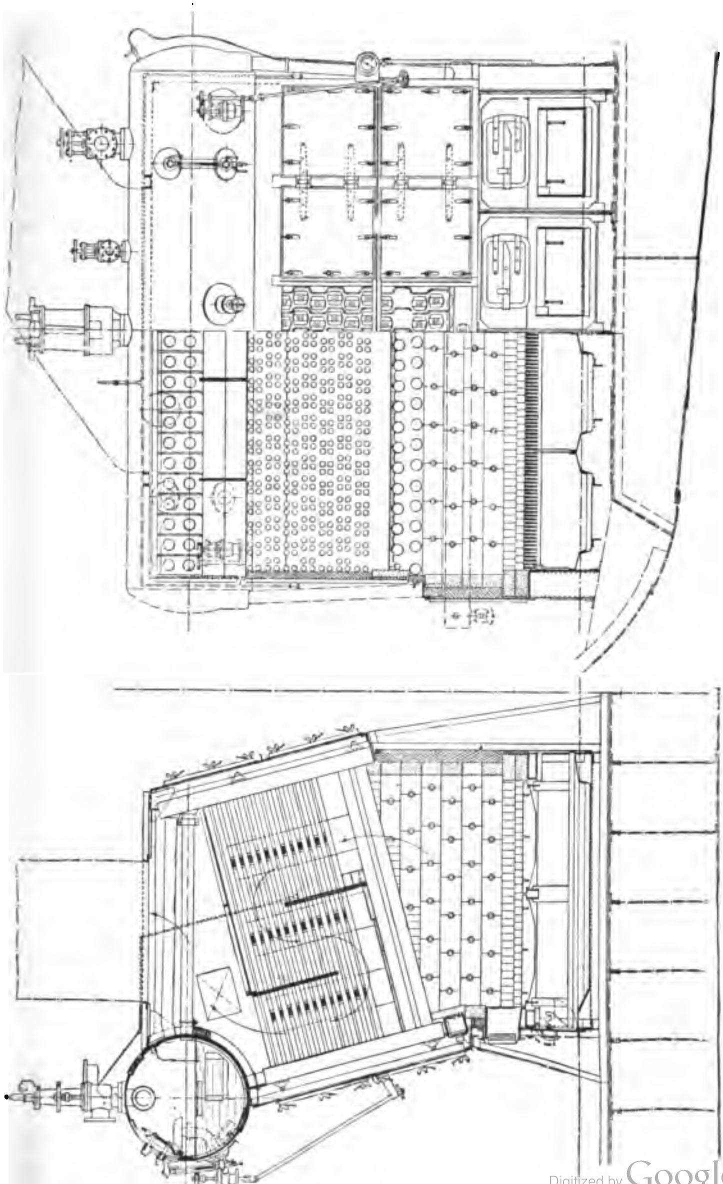


FIG. 61.—Babcock & Wilcox marine boiler.

the upper ends of the front headers by short tubes, is a horizontal steam and water drum of ample dimensions. The upper ends of the rear headers are also connected to this drum by horizontal tubes, each section of tubes being thereby provided with an inlet and outlet for steam and water. Placed across the bottom of the front headers, and connected thereto by short tubes or nipples, is a forged steel box of square section constituting the mud drum. This box being situated in the lowest corner of the bank of tubes, forms a blow-off connection or sediment box through which the boiler can be completely drained.

The furnace gases pass under the firebrick roof, located over the front portion of the second row of tubes, to a high combustion chamber at the rear, in which they are thoroughly mixed and burned before entering the bank of inclined tubes forming the heating surface. The furnace is built either of ordinary firebricks carefully fitted together, or of light fire tiles, which, by a special arrangement, are bolted to the side plates. The whole is encased in an arrangement of thin plating fitted with fire-refractory material which is so effective in reducing radiation losses that the outside of the casing is quite cool.

The mixture of water and steam formed inside the inclined tubes rises to the high rear end, and flows through the uptake headers and horizontal return tubes to the steam and water drum, being replaced by water flowing directly from the bottom of the steam and water drum downwards through the front headers into the tubes. By this means a rapid circulation is maintained as in the land type. On entering the drum, the steam and water impinge against baffle plates, which cause the water to be thrown downwards, while the steam separates and passes round the ends of the baffle plates to the steam space from which it is taken by a perforated pipe to the stop valve. The steam and water drum is also fitted with wash plates to prevent undue movement of the water in it when the ship is rolling.

7. The Stirling Boiler, Land Type.—There are three standard designs of Stirling boiler. In very small sizes the boiler has two steam drums and one mud drum ; in intermediate

sizes there are three steam drums and one mud drum; in the largest sizes the boiler has three steam drums and two mud drums. Fig. 62 shows the five-drum type arranged for hand firing and delivering saturated steam. The three upper steam drums are supported by brackets carried on steel beams which in turn rest on steel columns. These columns are built entirely

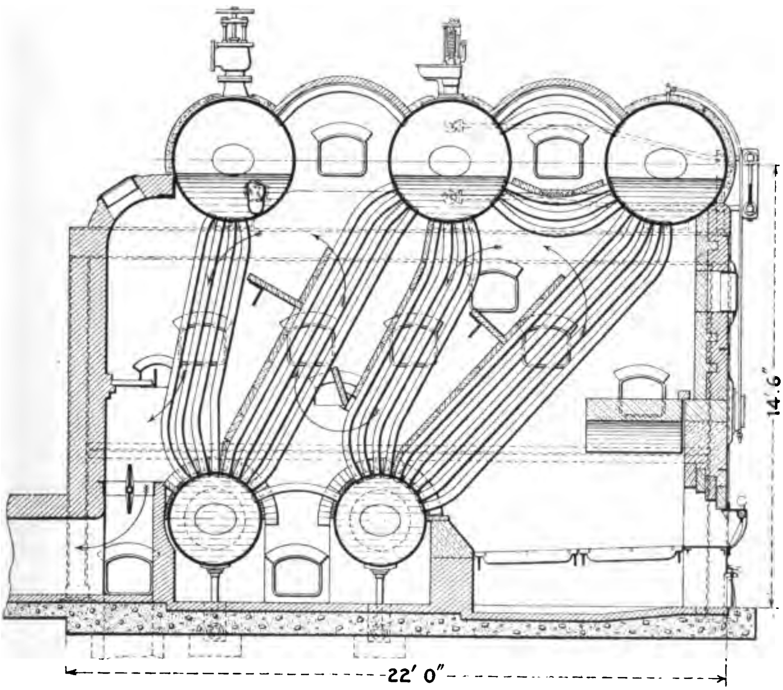


FIG. 62.—Stirling boiler, five-drum land type.

independent of the brickwork (as is usual with all types of water-tube boilers) so that this may be removed or replaced without disturbing the boiler or its connections. The mud drums are suspended by the tubes expanded into them, and not being in contact with the brickwork, are left free to accommodate themselves to any movement from expansion and contraction.

The drums are connected by four sets of vertical tubes $3\frac{1}{4}$ inches diameter, while water-connecting tubes are provided between the first and second steam drums and between the two mud drums. All the three steam drums are connected together by means of steam-connecting tubes expanded into the shells above the water level in the same manner as the main tubes. Suitably disposed firebrick baffle tiles are arranged between the banks of tubes to direct the furnace gases into their proper course.

Each drum is fitted with one manhole for cleaning and inspection purposes and the drum interiors are perfectly clear, there being no baffles, stays, etc., as shown in Fig. 62.

The feed water entering the rear top drum through the feed check valve passes into a feed distribution box which extends the whole length of the drum. By means of this box the feed is distributed over the whole width of the boiler, every tube receiving a proportion of the entering feed water. The feed water passes down the rear bank, up the third bank, down the second bank, and up the front bank of tubes. Here the steam formed during the passage up the tubes disengages and passes through the upper or steam circulating pipes into the middle steam drum, whilst the water passes through the lower or water-circulating tubes into the middle drum. This water again joins the main circulation and passes down the second bank and up the front bank of tubes, continuing its former course until it is all evaporated.

It will be noticed from Fig. 62 that there are no water-circulating pipes connecting the middle and rear steam drums and that the feed water is, therefore, compelled to pass down the rear bank of tubes to the mud drum. Should there be any scale-forming matter in the feed water, most of the deposit is precipitated in the rear mud drum which is not exposed to the intense heat of the furnace. Should any solid matter pass the third and second bank of tubes, there still remains another mud drum which collects any residue, and practically only pure water enters the front bank of tubes above the fire. The heaviest scale therefore forms where the temperature is lowest, *i.e.* in the rear mud drum, and some in the rear bank of tubes,

but the temperature here is so low that baking of the scale to a flinty hardness is obviated, and the deposit remains comparatively soft and easily removed, unless neglected for a long time.

The steam is withdrawn from the *rear* steam drum, and in this drum the circulation is downwards, there being no tendency to priming; in addition the steam from the front drum passes through the hot circulating pipes which dry it before the stop valve is reached.

The firebrick arch built above the firegrate in front of the front bank of tubes is a good feature because it absorbs heat from the fire, becoming an incandescent surface; this assists combustion of the gases given off from the coal (see Art. 7, Chap. III.), and reduces the risk of chilling the boiler by an inrush of cold air when the furnace doors are opened. The furnace gases flow in the opposite direction to the water in the boiler. After combustion takes place in the furnace, the gases pass up the front bank, down the second, up the third, and down the rear bank of tubes to the chimney. By this means, as in the Babcock boiler, the hottest gases are in contact with the tubes containing the hottest and cleanest water, and the coldest gases in contact with the tubes containing the coldest water, which conduces to durability and efficiency.

Boilers of the type shown in Fig. 62 are constructed to evaporate 30,000 pounds of water per hour; beyond that evaporation the Stirling Company recommend chain-grate stokers, in which case their largest boiler evaporates 40,000 pounds per hour with cheap inferior coal. Fig. 63 shows this type fitted with chain-grate stoker and integral superheater, while the small three-drum type is shown in Fig. 64.

8. The Stirling Boiler, Marine Type.—This boiler which is a modification of the land type, is shown in Plate III. It is of the four-drum type, with three upper steam drums and one mud drum, and differs from the five-drum type shown in Fig. 62 in having water-connecting pipes between the middle and rear steam drums. The marine boiler illustrated has tubes $2\frac{1}{4}$ inches diameter and is fitted with an integral superheater between the front steam drum and the middle steam drum. The superheater consists of one upper drum and one lower drum connected

together by a nearly vertical bank of tubes. The upper drum is fitted with a diaphragm (not shown), and the steam entering through the steam connecting pipes from the middle steam drum on one side of it, passes down half the tubes to the lower drum, and thence returns through the remainder of the tubes

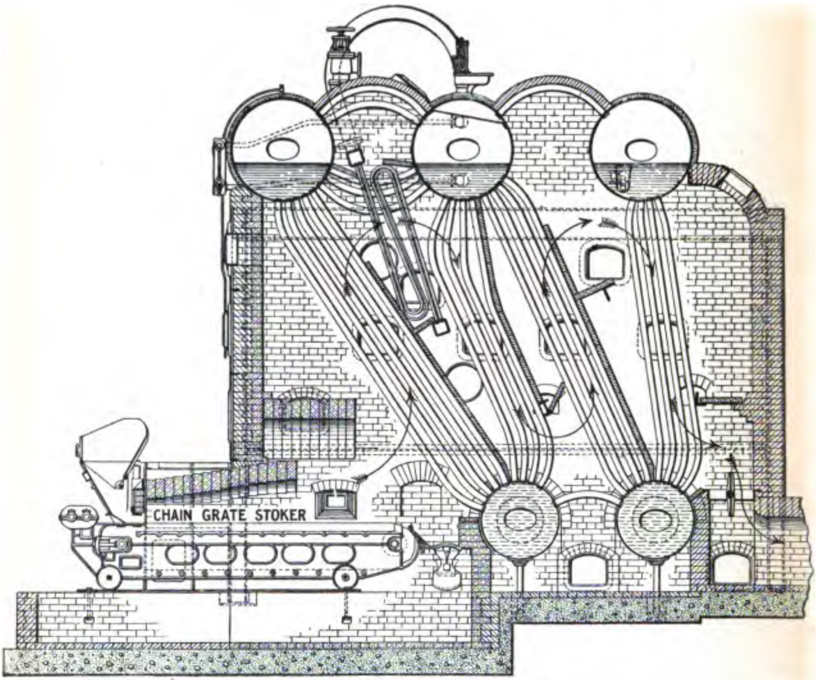
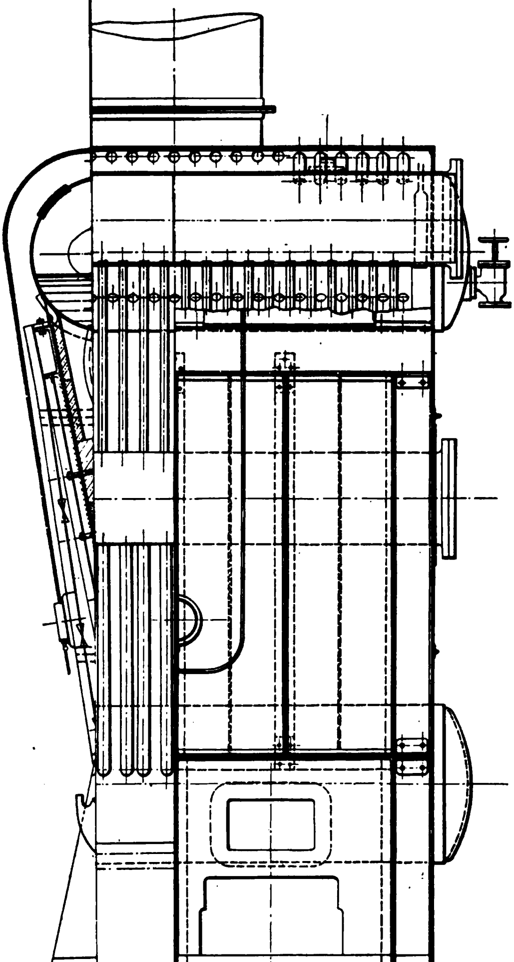


FIG. 63.—Stirling boiler, fitted with chain-grate stoker and integral superheater.

to the other side of the top drum on which the main stop valve is mounted.

9. Clarke-Chapman or "Woodeson" Boiler, Land Type.—This boiler (see Plate No. IV.) is made up of a number of sections, each consisting of a horizontal cylindrical steam drum at the top, a horizontal cylindrical water drum at the



le, $\frac{1}{4}$ in. = 1 ft.

bottom, and a number of groups of straight tubes expanded into flat discs on the steam and water drums. Each steam drum is connected to its neighbouring steam drum by water-circulating pipes below the water line and steam circulating pipes above the water line, and the water drums are connected in a similar manner as shown.

On the top of the steam drums are arranged a series of manholes, one immediately over each group of tubes, and any

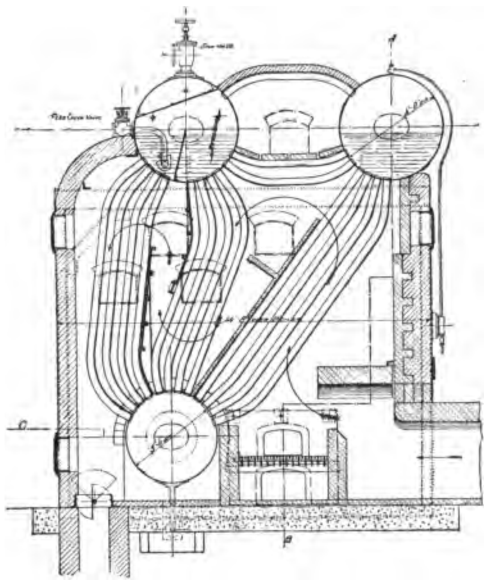


FIG. 64.—Stirling boiler, three-drum type for firing with waste gases from furnaces.

tube can be withdrawn and replaced through the manhole above the particular group in which the tube is situated. A large steam dome is arranged transversely over the three steam drums to ensure dry steam being obtained.

The feed water enters the rear steam drum at the point furthest from the fire, and the water flows down the vertical rear tubes in contact with the coolest gases, precipitating any

mud or dirt in the bottom rear drum. From the bottom rear drum, the water flows up the tubes in the front sections into the front and middle steam drums where the steam is liberated and from which it passes into the steam dome. By this arrangement all the tubes in the front sections exposed to the

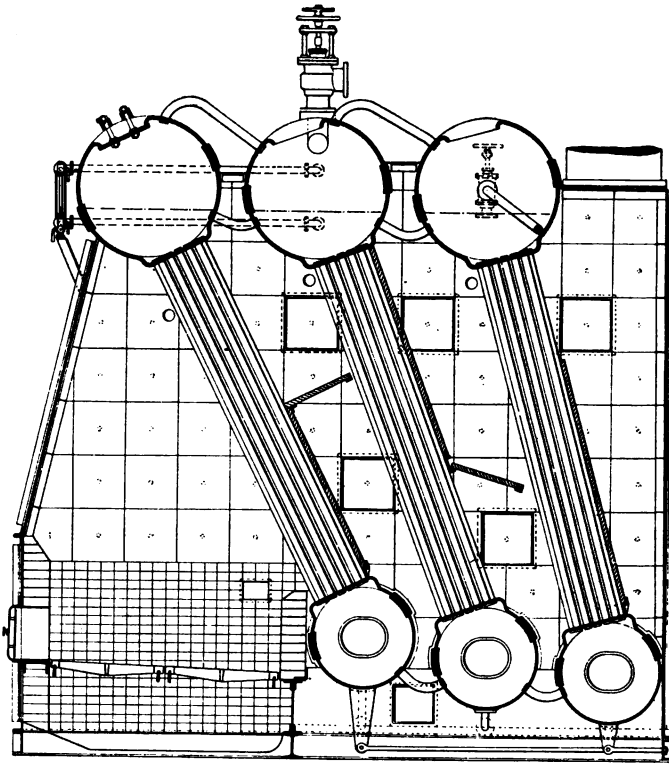


FIG. 65.—Clarke-Chapman marine boiler (Woodeson's Patent).

greatest heat contain practically pure water, as in the case of the Stirling boiler previously described (Art. 7).

The firegrate is arranged across the boiler immediately in front of the front sections of tubes, the furnace gases travelling upwards among the tubes of the front sections, over the fire-

brick baffle arranged between the sections, downwards among the tubes of the middle section and upwards among the tubes of the rear section and then off to the chimney.

When a superheater is fitted it is arranged between the front and middle sections of tubes as shown in Plate IV., the steam being taken from the steam dome through the superheater tubes to the stop valve.

The whole boiler is suspended from steel cross girders which are carried on vertical steel columns, and in this respect is similar to the Babcock & Wilcox and the Stirling boilers (see pp. 185 and 189).

10. Clarke-Chapman Boiler, Marine Type.—This boiler as shown in Fig. 65 is a modification of the Land type, the essential differences consisting, as usual, in the absence of brickwork and shorter tubes in order to reduce head room. The boiler shown in Fig. 65 is designed to give saturated steam, the steam being withdrawn through the stop valve on the middle steam drum. When superheated steam is required from the marine boiler, the integral superheater is placed immediately below the uptake to the funnel, there being only two steam and two water drums as indicated in Fig. 66.

11. Nesdrum Boiler (Hornsby's Patents).—This boiler now manufactured by Messrs. Richardsons, Westgarth & Co., Ltd., is illustrated in Fig. 67. It consists of a number of short cylindrical headers with spherical or dished ends. Each pair of headers is connected together by means of a nest of straight tubes, and each nest of tubes, with its bottom header acting as a water drum and its top header as a steam and water drum, constitutes a separate boiler in itself. The top headers are connected together by steam-circulating pipes or nipples above the water line and water-circulating pipes below the water line, the bottom headers being similarly connected as shown. The top headers are also connected to the main steam drum by expanded pipes in both the steam and water spaces. The feed water is admitted into the rear top headers behind the main steam drum and flows down the vertical nests of tubes exposed to the coolest gases depositing the bulk of the scale-forming

matter as mud in the bottom headers from which it is periodically blown out. The front sets of tubes being exposed to the hottest gases, act as steam generators, the water and steam ascending from their bottom headers into the top headers where the steam is liberated and passes through the steam-circulating

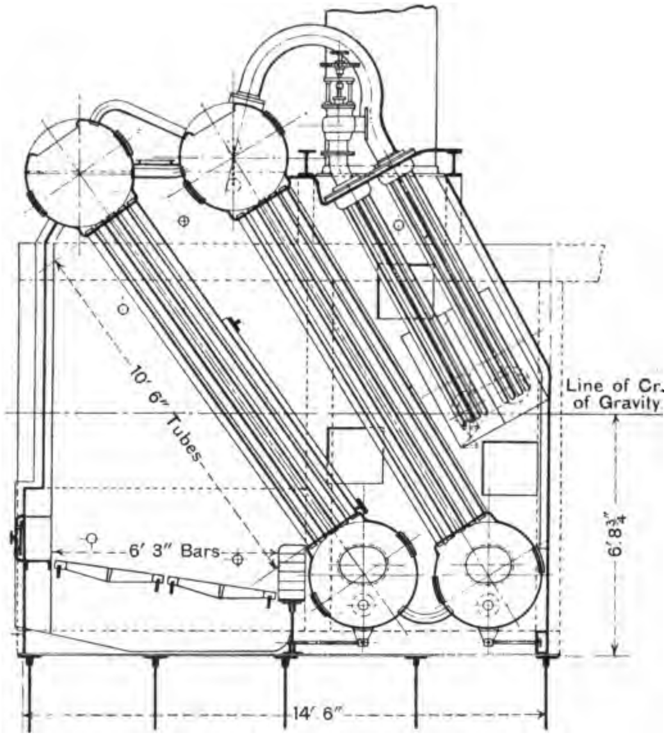


Fig. 66.—Clarke-Chapman marine boiler, fitted with integral superheater.

pipes to the main steam drum. Each header is fitted with a manhole through which any tube can be replaced or withdrawn. The top headers and main steam drum are slung from overhead girders, and the remarks already made on p. 185 apply equally as well to this as to other water-tube boilers.

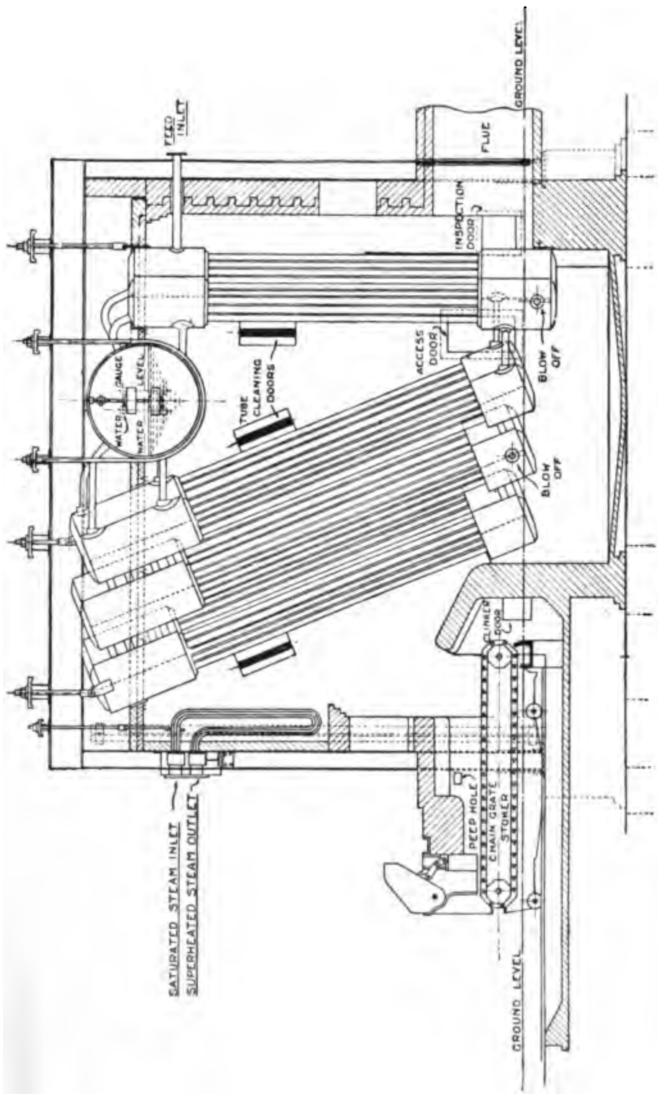


FIG. 67.—Nesdrum water-tube boiler.

The tubes are staggered in such a way as to break up the gases when passing through the first row of nests of tubes, after which they re-unite, and are then broken up again by the second row of nests of tubes, and so on. The only baffle plates required are placed at the back of the rear vertical nests of tubes, in order to deflect the gases in the proper direction before entering the flue. The boiler tubes are therefore not obstructed by baffling tiles, so that the soot can easily be swept off the tubes by means of a steam jet when working, and by brushes when the boiler is laid off for periodic cleaning and inspection.

The boiler illustrated is fitted with a special type of superheater which has proved very reliable in practice. The makers do not recommend flooding (Art. 21, Chap. VIII.) of their superheaters at any time, especially if the feed water is hard or if any oil is likely to be present in it, because, if any scale is formed or any oil passed through with the steam, the tubes are liable to burn out, whereas if flooding is not resorted to, the tubes always keep clean inside. They find, by experience, that flooding is quite unnecessary when raising steam. The superheater, it will be noted, is placed in front of the combustion chamber, above the furnace grate; so it only receives the radiant heat from the gases, and therefore does not become coated with soot or other deposit on the outside of the tubes.

The boiler is made in very large sizes, the largest, installed at Charing Cross, in the West End and City Electricity Supply Co.'s Station, evaporating 100,000 pounds of water per hour under normal working conditions.

12. Suckling Boiler.—This boiler is also of the upright water-tube type, and consists of two cylindrical steel drums, one placed vertically above the other and connected together at each end by two vertical pipes of large diameter. The upper, or steam drum A (Fig. 68), is placed horizontally, and the lower, or water drum B is given a considerable slope upwards towards the front of the boiler, the blow-off cock being fitted at its lower end as shown. Of the two vertical connecting pipes C and D, the front one, C, is in the direct path of the hot gases leaving the furnace, which is arranged externally to the main setting; it thus acts as an uptake for the water, whilst the

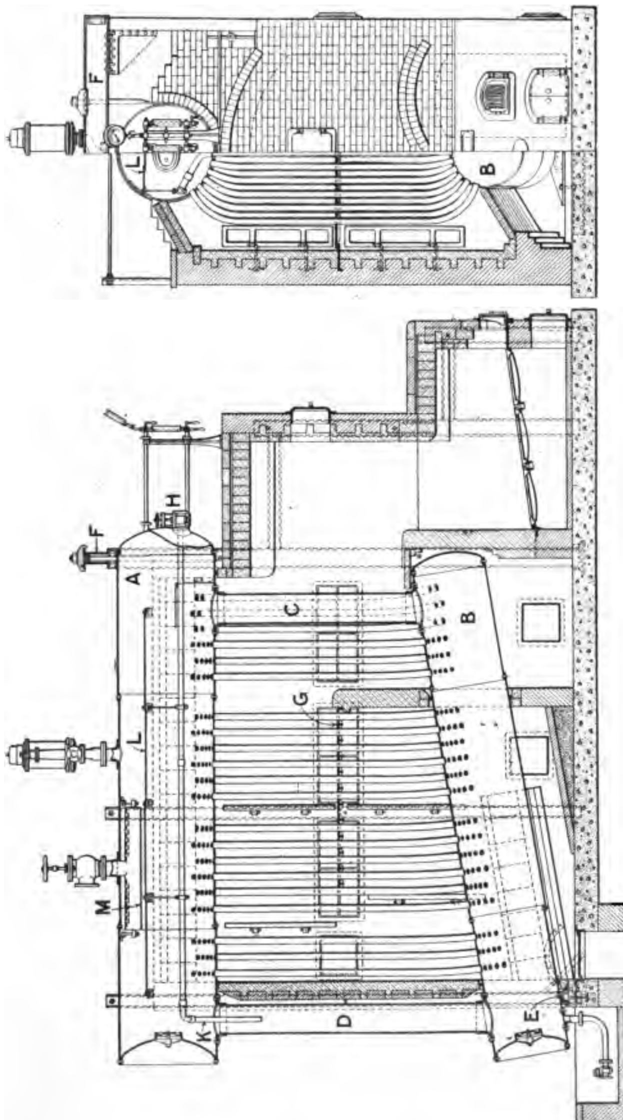


FIG. 68.—Suckling water-tube boiler (Messrs. Thomas Beeley & Son, Ltd.).

pipe D at the back end of the drums, being outside the setting, acts as a downcomer. The two drums A and B are further connected by groups of vertical steel tubes $3\frac{1}{4}$ inches external diameter, constituting the main portion of the heating surface; these tubes are arranged in staggered rows so as to obtain full benefit from the gases flowing through the boiler.

The ends of the water tubes are slightly curved in order to bring them normal to the drums into which they are expanded. The curvature given to the tubes also allows the drums to be made cylindrical, without any flats for tube connections, and further, makes provision for expansion between the drums.

The two drums with their connecting pipes and tubes constitute a complete section. Each section is a complete unit in itself, and may be duplicated according to the amount of heating surface required.

The bottom water drum B rests on a cast-iron cradle E near its rear end, this cradle being supported by a steel girder bedded in the concrete foundation. The front end of this drum is carried by the uptake pipe C riveted to the steam drum A, which latter is supported at the front end by a steel stirrup from the cross girder F. The back end of the top drum is carried by the downcomer tube D, and it will be seen that both drums are free to expand lengthwise, and that no weight is carried by the water tubes or brickwork setting.

A manhole is provided in the dished end of each drum to give access to the tubes for cleaning and inspection purposes. The tubes are so arranged that individual tubes can be easily taken out and replaced without interfering with adjacent ones; further, no extra head-room is required for this operation since the tubes are passed in from the sides of the boiler.

The external furnace, lined with firebrick, is of sufficient size to ensure a thorough mixing of the gases and complete combustion before they come into contact with the comparatively cool heating surface (Art. 1, Chap. III.). On leaving the furnace, the gases take an upward course, and then travel along between the steam drum A and a series of horizontal cast iron baffle-plates G extending across the setting until they reach the back end of the boiler, whence they return beneath

these baffles towards the front. On reaching the cross-wall the gases again turn backwards and pass along underneath the water drum B to the damper opening.

The feed water enters the boiler at the front end of the top drum through the feed check valve H, and passing through the internal pipe K is delivered into the downcomer tube D at the back of the boiler. The slope given to the bottom drum B allows the sedimentary impurities to be deposited at its back end (near the blow-off attachment), and also ensures a forward and upward travel of the water and steam along the bottom drum to the uptake C; a continuous and rapid water circulation is obtained by this means.

Each water tube draws its own unrestricted supply of water from the water drum B, and the steam generated is liberated in the steam drum A in a series of small jets from the tubes along the whole of the ample water surface in the steam drum; there is, therefore, no violent ebullition or frequent fluctuation of the water-level, and consequently any tendency to priming is greatly reduced. The further provision of a perforated diaphragm L above the water-level in the steam drum in addition to the usual anti-priming pipe M also assists in the supply of dry steam. A superheater is only fitted when greater economy in coal consumption is desired.

The first boiler of this type has now been working for seven years with complete immunity from trouble with leaky joints, and no water tubes have, as yet, required renewal. Another great advantage of this boiler is the facility for cleaning and the ease with which it can be opened out for inspection.

13. Niclausse Boiler.—This boiler (Fig. 69) is very largely used in the French and British navies, as well as in land practice. The tubes are 12 feet 4 inches long and 84 mm. internal diameter, and each tube consists of two parts, viz. an outer tube, in which the water is evaporated, and an inner tube, along which the water to be evaporated is carried to the back end of the outer tube. The inner tube also acts as a feed-water heater. The outer tube is made of a solid-drawn mild steel tube, and is closed at the back with a screwed cap. The front end is staved, and has two coned surfaces which fit into cor-

responding coned holes in the front and back walls of the header. Between these two cones, two oblong ports are cut out of the tube which communicate with the compartments in the header for the descending water and rising steam respectively.

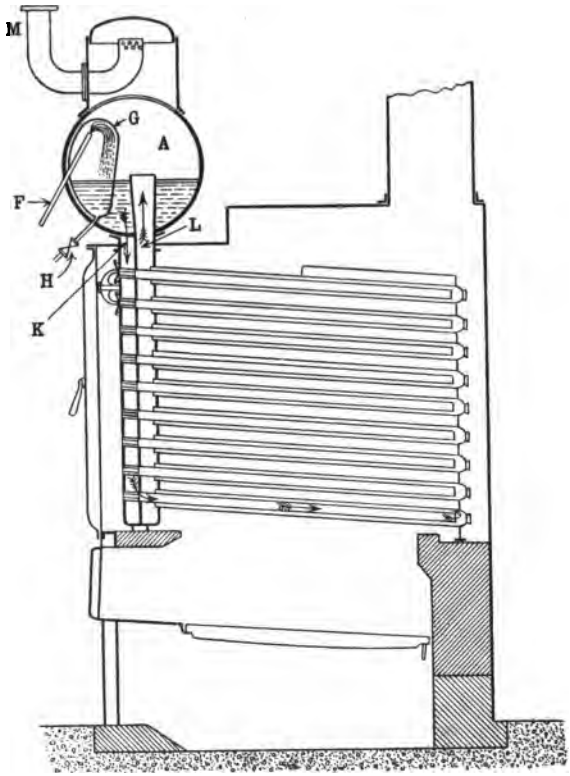


FIG. 69.—Niclausse boiler.

The inner tube, when in use, being full of water and in direct communication with the outer tube through its back end, is exposed to equal external and internal pressure, and is, therefore, made of thin sheet steel. The front end opens into the front division of the header, and is connected by one strap to a

screwed head, so designed as to screw into and close the front end of the outer tube. The details of the tubes showing their attachment to the header are shown in Fig. 70.

The headers to which the tubes are connected are made of solid-drawn steel tubes of square section, and are placed vertically in the boiler. They are constructed with a central internal diaphragm fixed at right angles to the tubes, dividing the header into two compartments K and L (Fig. 69), the front one, K, for the descending water, and the other for the ascending steam and water. In the front and back walls there

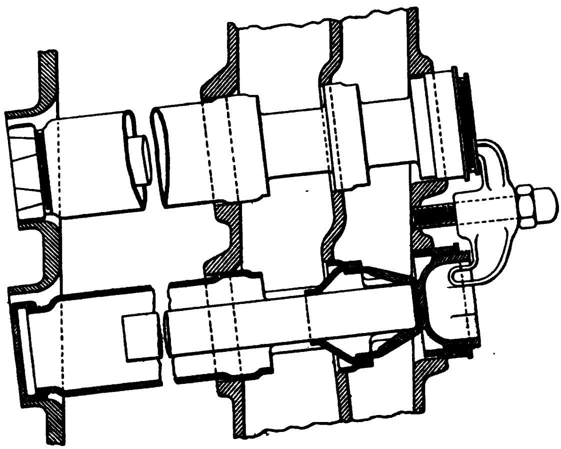


FIG. 70.—Details of attachment of tubes to header of Niclausse boiler.

are machined coned holes, into which the corresponding cones on the tubes are inserted. The tubes pass freely through the diaphragm, which is placed so as to come between the two ports in the outer tube. The holes in the header for the tubes are staggered, so as to prevent straight passages for the furnace gases to pass through.

The Niclausse joint, which is one of the essential parts of this boiler, is made by pushing in the tube until the cones on it are pressed well home into the coned holes in the header; a metal-to-metal joint is thus obtained. The coned holes in the headers are machined to gauge, and any tube will fit into

any hole and make a steam and water tight joint. The efficiency of these tube joints is ensured by making the back cone on the tube somewhat flexible, and it is so designed that it comes into contact with the coned hole in the back wall of the header before the front cone on the tube, which is rigid, does so with the coned hole in the front wall. On further pushing in the tube the joint at the back of the header yields until the front cone on the tube makes a good joint. In this manner steam-tight joints are obtained.

The inner faces of the tube cones are approximately of the same diameter. It therefore follows that since the internal pressure acts on two nearly equal surfaces in opposite directions, the tendency for a tube to be pushed out of the header by pressure is small, much smaller than the friction of the two joints, and it is only necessary to provide a bridge piece, as shown in Fig. 70, to prevent the tubes being loosened by some abnormal cause, such as shocks or vibrations.

The top end of each header is cylindrical and machined all over, and is tapered for a depth of about 2 inches. This tapered portion forms the joint between the header and the steam drum when it is inserted into a corresponding coned hole in the bottom of the steam drum.

In the older type of boiler each header was made of malleable cast iron, and was fitted with a separate blow-off, but in the latest design the separate blow-offs have been superseded by a horizontal pipe connecting all the headers and provided with a common blow-off attachment. The number of headers depends on the size of the boiler, and varies from 3 to 15.

The cylindrical steam and water drum A, to which the headers are attached (Fig. 69), is made of steel plates riveted together, its length depending on the number of headers. The bottom of the drum is strengthened by a thick steel saddle plate, which is riveted to the drum and to which the headers are secured. Inside the drum there is a sediment box or "lime depositer" G, made of thin sheet plate in the form of a trough, into which the feed water is injected by the pipe F. A large proportion of the impurities contained in the feed are thus deposited at the bottom of the trough, whence they can be

removed periodically by means of the blow-off cock H. The water-level in the "lime depositer" is a little higher than in the drum, and the feed flows over a lip and is distributed along the whole length of the drum.

The back compartment L of each header inside the drum is surmounted with a funnel, the object of which is to prevent the feed, which flows down the front compartment K of the header, from being impeded by the upflow of steam. The drum is provided with a dome containing an internal collecting and anti-priming pipe which takes the steam from the upper part of the dome through the pipe M to which the main stop valve is attached. A manhole is also provided in the drum in a suitable position.

The water circulation is as follows: The water from the drum descends by the front compartment K of the headers and passes into the inner tubes, whence it returns, from the back end, by the annular spaces between the inner and outer tubes. The steam is formed in these annular spaces, and flows into the back compartment L of the headers, from which it rises into the drum A.

The first two rows of tubes receive heat from the fire chiefly by radiation; the remaining tubes receive their heat by conduction from the hot gases. The number of rows of tubes is such that the best practical results are obtained when burning about 16 pounds of coal per hour per square foot of grate, but it is stated that the rate of combustion can be forced up to 50 pounds per hour per square foot of grate without detriment to the tubes or straining of the boiler.

In the latest type of Niclausse boiler an alteration has been made in the method of water circulation, which greatly improves the working of the boiler and from which excellent results have been obtained. With this new arrangement the tubes have to be cleaned internally far less frequently, and the life of the lower tubes is greatly prolonged. The general design of the boiler is not altered, but a vertical diaphragm is placed in the drum, as shown in Fig. 71, separating the front portion of the drum, which contains the feed water after it has passed the lime depositor, from the rear portion containing the water from

which steam is being generated. This diaphragm is so arranged that the feed water descends the front compartment of about four out of every five headers to a point more than halfway down, where there is a horizontal diaphragm in each header, as shown in Fig. 71. These headers are called the *purifying* headers, the remainder being called the *feed* headers. The feed only enters the top five or six rows of tubes of the *purifying* headers; these tubes act as feed-heaters, and the lime salts not caught by the lime depositor in the steam drum are deposited

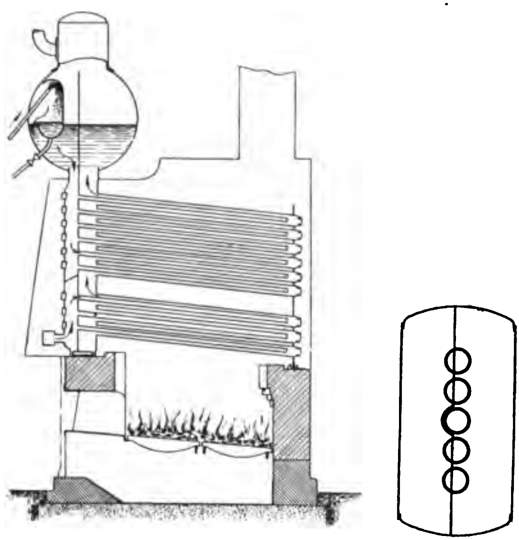


FIG. 71.—Latest type of Niclausse boiler.

in them. These deposits are not detrimental because the upper rows of tubes are not exposed to the radiant heat of the furnace; they therefore do not harden and are comparatively easily removed.

The feed, after passing through these upper tubes, contains no scale-forming material and its temperature has also been raised to that corresponding to the steam pressure. This hot purified feed then rises *viâ* the back compartments of the purifying headers into the rear portion of the drum; thence it

flows down the whole length of the front compartment of the feed headers, circulating in all their tubes, and, by means of a horizontal pipe connecting the bottom ends of all the headers (the blow-off is attached to this pipe as already mentioned), also circulates in the lower tubes of all the *purifying* headers. By this means all the lower tubes of the boiler are fed by pure water, whose temperature is that due to the pressure, and consequently the evaporation in these tubes is greatly increased. The boiler can therefore be forced to a much greater extent than was possible in the older type (Fig. 69), and there is no difficulty in burning 50 pounds of coal per hour per square foot of grate area.

The casing of the boiler is arranged so that the boiler is supported on the same principles as other types of water-tube boilers, as already mentioned on pp. 185 and 189. The height of the first row of tubes above the firegrate is such as to give a suitable size to the combustion chamber depending upon the particular class of fuel to be used. The combustion chamber is lined with firebrick.

The back ends of the tubes are supported by vertical cast-iron plates, perforated with holes, in which the tubes rest freely. These cast-iron plates also protect the back of the boiler, and, especially, the caps closing the back ends of the outer tubes, from the action of the flames.

14. Vogt Boiler.—This type of water-tube boiler is extensively used in the United States. It is built with one or more horizontal steam drums A extending its full length (Fig. 72). These drums provide a storage capacity for steam which prevents sudden fluctuation in pressure; they also provide a very large liberating surface which allows the steam to escape freely through a comparatively shallow body of water and secures dry steam.

The heating surface consists of three rows or banks of horizontally inclined tubes, B, C, and D, connected by cross drums, and one vertical bank E in the rear of the boiler connecting the steam drums with the mud drum F. The feed water enters the steam drums either through their front or rear ends; it is distributed directly above the vertical

bank of tubes in the rear and passes down to the mud drum,

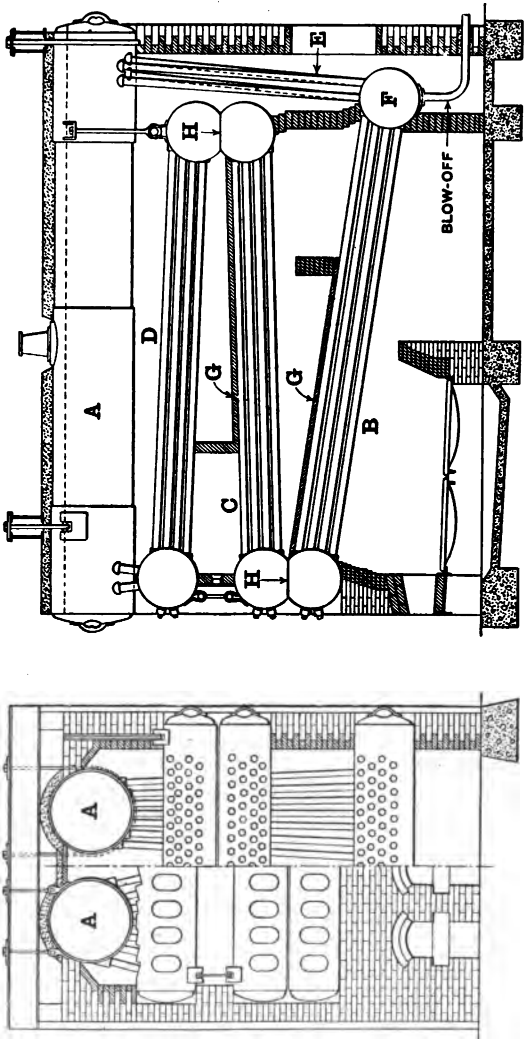


FIG. 72.—Vogt water-tube boiler.

then through the three banks of tubes, previously mentioned, and back into the steam drums, so completing the circuit.

One of the great advantages claimed for this arrangement is its flexibility, which allows freedom for the expansion and contraction of the tubes. Then, as the water has to pass back and forth through the hot gases three times before again reaching the steam drums, very rapid circulation results. This tends to prevent the formation of scale in the tubes, and this fact, together with the free expansion and contraction, accounts for tubes rarely having to be renewed. The entering feed water passes directly to the rear tubes, causing the coldest water to come in contact with the coldest gases, so that the temperature of the flue gases leaving the boiler is a minimum. It also causes the feed water to be heated to a temperature high enough to separate scale-forming substances before reaching the mud drum, where they are deposited and blown off; any scale-forming substances are thus largely prevented from reaching the horizontally inclined tubes.

The baffle tiles G compel the hot gases to pass backwards and forwards three times along the length of the boiler before passing down over the rear vertical tubes to the chimney.

The tubes are all straight and of the best lap-welded or cold-drawn steel, and are expanded into reamed holes in the cross drum plates. The upper and lower sections of each twin cross drum are riveted together with a plate H in the centre between the two halves; this centre plate is perforated, allowing the water and steam bubbles to pass freely from the lower to the upper half of the twin drum. This plate which unites the two drums into one is the only straight plate used in the construction, but being perforated, the same pressure exists on each side of it, so that there is no tendency for the plate to be deflected out of its natural flat position.

The boiler is suspended from a framework of channels and I beams, no part of the boiler resting on the brick walls. As shown in Fig. 72, each top steam drum is suspended by two eye-bolts and one stirrup; these have threaded ends and nuts and pass through steel washers on the tops of channel girders, which facilitates the levelling of the boiler. The eye-bolts are secured to the drums by steel pins and loops riveted to the drum shell. The front twin drum is suspended from the upper front

cross drum by two links with pins and pressed steel loops riveted to the drums. The upper twin drum in the rear is suspended by eye-bolts from pressed steel brackets riveted to the top steam drums. The mud drum is suspended from the steam drums by the vertical bank of tubes in the rear, which being expanded at the top and bottom supports the drum. The entire arrangement of suspension and supports for the several sections of the boiler makes it very flexible.

15. Casey-Hedges Boiler.—This boiler is also used with considerable success in the United States. Simplicity of construction is one of the chief features in the design. The boiler consists of one or more steam and water drums A (Fig. 73), having two wrought steel headers or water legs B and C, one at each end, each header consisting of a handhole plate and a tube plate. The headers are thoroughly braced with hollow stay bolts of large diameter. The front water leg B is 12 inches wide at the top, so that there is no restriction of area at this point, and the steam and water can freely circulate. The rear leg C is 10 inches wide at the top, the lower portion being increased in width to meet the inclination of the lower bank of tubes at right angles, and forming a large settling chamber at this point.

The tubes are divided into two banks, the upper bank and the drum being inclined 1 inch to the foot and the lower bank 2 inches to the foot; the lower tubes being the hottest, their inclination should be the greatest. It is claimed that this construction permits of a large area at the rear of the tubes and allows for complete expansion of the hot gases at this point, the area decreasing as it reaches the front end, where the gases are cooler. The upper baffle D consists of special V tiles. The design of the baffles is such that the passages for the gases may be decreased or increased to suit the fuel and draught conditions. The lower bank of tubes is completely surrounded, with the exception of a portion at the rear end, with a fire tile casing E, giving a *Dutch oven* effect to the furnaces, ensuring complete combustion of the gases and practically a smokeless boiler with suitable fuel.

The water circulation is down the rear leg, up the tubes into the steam and water drum. In order to provide an unrestricted area, the front header or water leg is 12 inches wide at the top,

allowing for a large outlet for the steam and water to reach the liberating surface. The steam outlet is at the front end of the

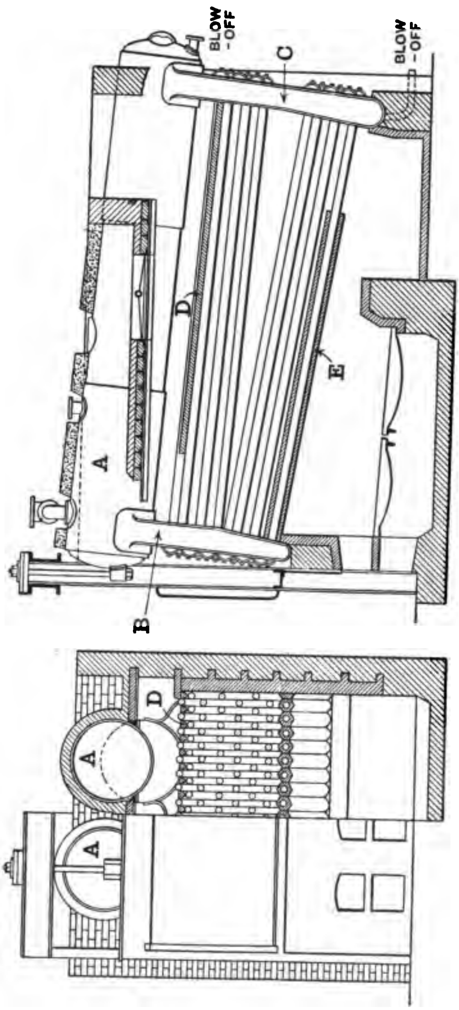


FIG. 73.—Casey-Hedges water-tube boiler.

boiler and is provided with an antipriming pipe, also a deflector or baffle plate, ensuring practically dry steam; the steam outlet

is about three-fourths the diameter of the drum away from the water-level.

The downward circulation is through the lower header or water leg which is 10 inches wide at the top. The lower part of this rear leg is swelled out as mentioned above, and forms a precipitating chamber for all solids that have not been deposited in the mud drum, the blow-off being tapped in the extreme bottom of the rear leg and enabling the boiler to be completely drained off through it.

The mud drum is located in the top steam drum, being 8 inches in diameter and running almost the full length of the steam drum. The feed water is admitted into the mud drum and most of the sediment is deposited therein; a blow-off extends from the rear of the mud drum out through the rear end of the steam drum, through which the sediment can be blown at intervals.

Each of the headers are stayed with hollow stay bolts so arranged that a steam blower can be inserted through them and all soot that might have collected on the tubes and tiling can be blown into the combustion chamber while the boiler is in operation, without admitting cold air into the setting.

To clean the interior of the boiler, there is provided in each header opposite the end of each tube a wrought-steel handhole plate that tightens under internal pressure and throws no strain on the handhole bolt. The tubes are easily inspected by taking off a few of the handhole plates in the bottom row, and as many as five tubes can be cleaned from one handhole opening in the front end.

The boiler is supported independently of the brickwork by means of hangers and steel girders, consisting of channel beams resting on I beam columns, which are attached to and form part of the boiler front. The rear header rests on heavy cast-iron columns, with saddles and rollers, allowing the boiler to expand and contract free from the masonry. The design of the front is such that any type of mechanical stoker can be attached without trouble.

16. Marshall Boiler.—This boiler is of a simple and efficient type and consists of one or more steel steam drums, according

to the size of the boiler, with a steel header, or water leg for carrying the tubes, riveted direct to the drum at each end as shown in Fig. 74. The connection of the water legs to the drum,

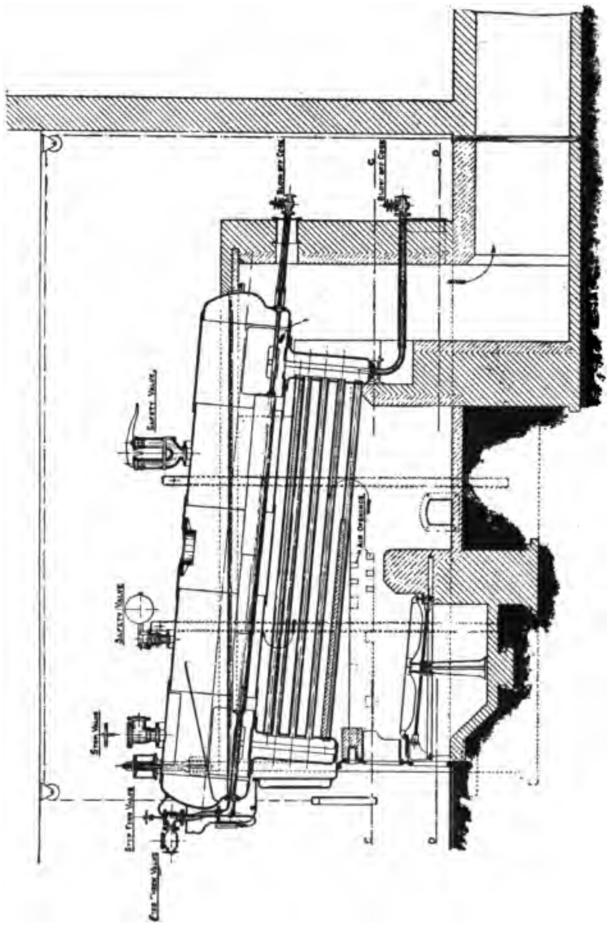


FIG. 74.—Marshall water-tube boiler.

or drums, is made with ample water way in order to provide for thorough circulation, and the water legs themselves afford a large cross-sectional area so that the boiler will stand forcing without any risk of the tubes being left bare.

The straight, solid-drawn steel water tubes which connect the two water legs are expanded into holes in the inner plates of the water legs, the outer plates having corresponding holes, slightly larger in diameter, fitted with lids opposite each tube end for inspection and cleaning purposes. The front and back plates of the water legs are tied together by hollow steel screwed stays, and when the boiler is set in brickwork, as in Fig. 74, a steam jet pipe can be passed through the stays in order to blow the soot off the outside of the tubes. The tubes are parallel to the drums and the boiler is set with an incline downwards from the front to the back end, being slung from an overhead cross girder at the front end, and resting on the brickwork setting just in front of the rear header or water leg as shown. The standard boiler of this type is constructed for a working pressure of 150 pounds per square inch.

Small Tube Boilers.

17. Thornycroft Boiler.—The early form of Thornycroft boiler (1885 design) is generally known as the "Speedy" type, since this was the first vessel in the British Navy of any importance to be fitted with the water-tube boiler. As will be seen from Fig. 75 there are in this type of boiler three horizontal drums. The two lower ones A and B on either side are water drums and the upper one C is a steam and water drum, to which the lower drums are connected by two large pipes D and also by two groups of small curved generating tubes, usually of from 1 inch to $1\frac{1}{4}$ inch diameter.

The two large pipes D, which are outside the casing of the boiler, are for the return of the unevaporated water which has been carried over from the generating tubes with the steam, and for the feed water entering the upper steam and water drum. By this arrangement there is a definite circulation which secures a constant flow of water over the heating surfaces. Both groups of generating tubes are bounded on either side by a wall of water tubes, each of which is formed by bringing two rows of the tubes together so as to follow the same curves in a transverse section of the boiler, except at the upper and lower ends, where they are

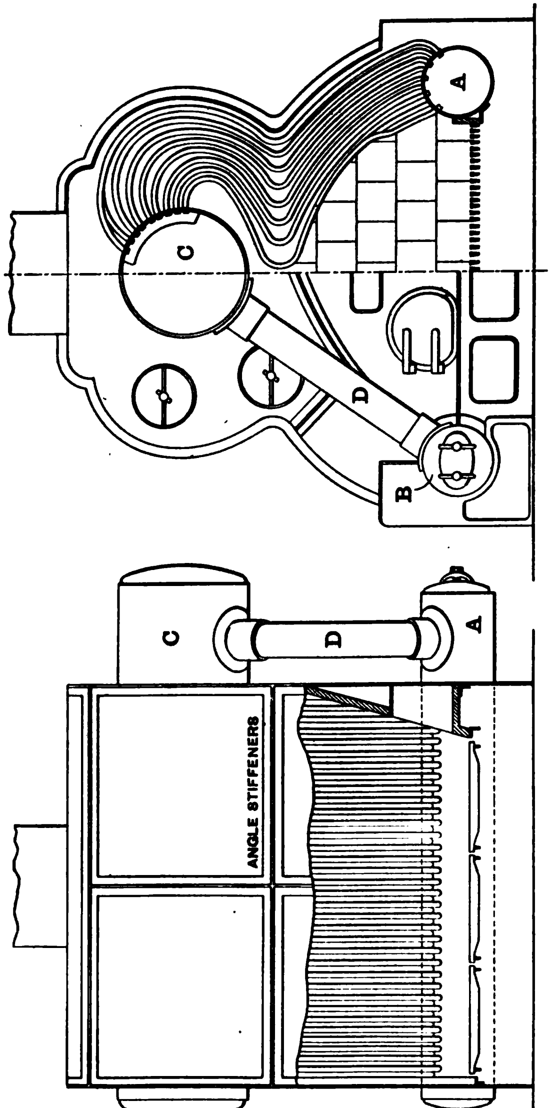


FIG. 75.—Thornycroft boiler, "Speedy" type.

opened out in order to enter the drums. Except in the case of the walls, there is between the tubes a space of about $\frac{1}{2}$ inch in a longitudinal direction (*i.e.* from front to back of the boiler), and the various rows are about two diameters apart as seen in a transverse direction, so that there is a space all round the majority of the tubes. The two groups of tubes thus form a pair of flues along which the hot gases pass on their way to the chimney.

In order that the hot gases may pass along these flues and out again into the uptake, the splaying or opening out of the wall tubes is prolonged, where necessary, to give the requisite area. The gases pass into the flues from the furnace, near the lower water drums, and after passing along the tubes, emerge through the two external rows on either side of the steam drum.

The rows of tubes next to the furnace are bent so as to form an arch over the firegrate, and after meeting in the middle are bent back so as to enter the steam drum at the requisite position as shown. The hot gases are by this means prevented from coming into contact with the lower part of the steam drum, so that no ebullition is caused in the water therein, and, should the water be low, the risk of overheating the top steam drum, which would otherwise be just over the fire, is obviated. This arrangement of bending the tubes also gives the greatest facility for expansion of the tubes that most need it, and by allowing all the tubes to expand freely, lessens the likelihood of their becoming strained or loosened where they are expanded into the drums.

A casing of asbestos and steel envelopes the whole boiler, the ends of the furnace and combustion chamber being formed of firebrick attached to a nearly vertical sloping plate. A space is left between this plate and the outer casing, in which air doors, opening inwards, are so placed that a current of air is made to pass over the brickplate and down to the ashpit. The ashpit is also provided with air doors to protect the stokers from a rush of steam in case of a bad leak.

"*Daring*" *Type.*—When torpedo boat destroyers were first built for the British Navy, Mr. (now Sir J. I.) Thornycroft introduced the "Daring" type of boiler (Fig. 76) so called from H.M.S. *Daring*, the vessel on which they were first fitted, his object being

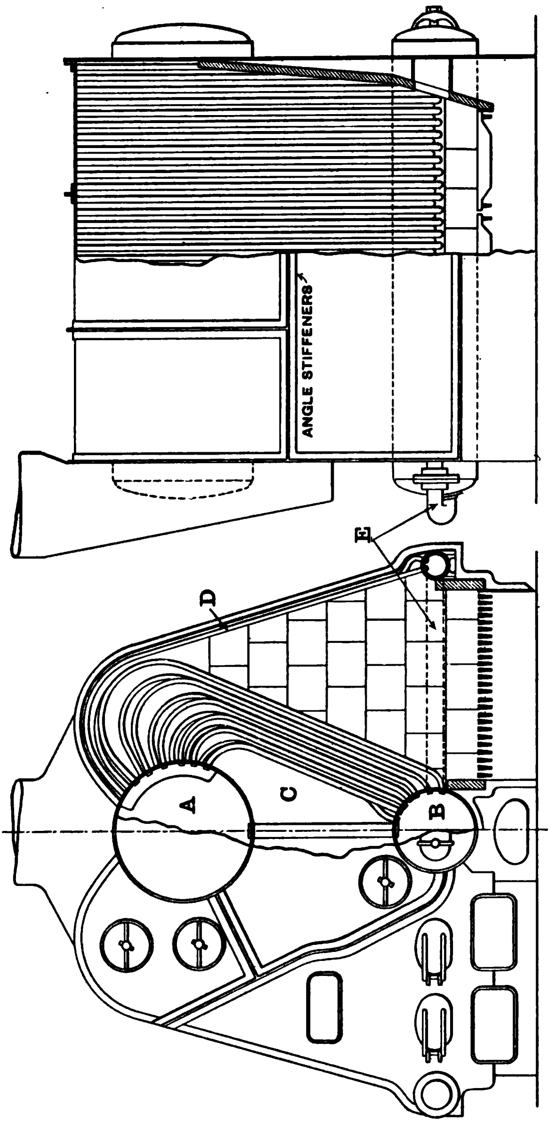


FIG. 76.—Thornycroft boiler, "Daring" type.

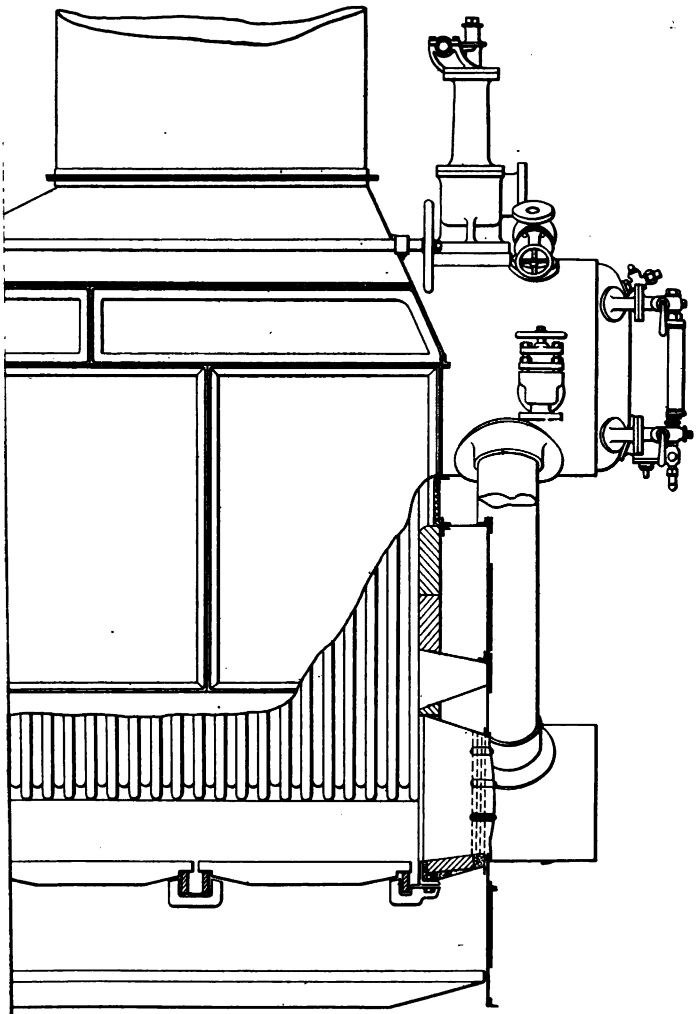
to secure as large a grate area as possible. In its original form this boiler has two parallel drums one over the other, the upper one A being the steam, and the lower one B the water drum. They are connected together by some eight or nine large vertical downtake tubes generally about 4 inches diameter, and also by two groups of curved generating tubes C, each bounded by a pair of water-tube walls D as in the case of the "Speedy" type previously considered; on each side of the lower water drum is a firegrate. The two fireboxes or combustion chambers are each bounded on one side by one of the two groups of tubes C, and on the other by a water-tube wall D formed as described above, the ends being closed by bricks and plates as in the Speedy type. These wall tubes are supplied with water by a pipe E connected with the water drum and bent round on the outer sides of the grates.

The gases pass through the firebox walls into the groups of tubes by the apertures near the lower drum and up along the tubes in a diagonal direction, emerging through the central wall of tubes into the heart-shaped central space under the steam drum. They then pass along to the back of the boiler into the smoke box and up the chimney.

Broadly speaking, the "Speedy" type is most suitable for moderate-sized units, especially if a large heating surface is required in comparison with the great area. The "Daring" type appears most convenient for large boilers, particularly when a large grate area is wanted.

The above designs of Thornycroft boiler have one common feature, namely, that all the steam generating tubes deliver their contents into the upper steam drum *above* the water-level. This arrangement, which affords a very rapid circulation and large evaporating area, combined with the baffle plates or grids which the steam has to pass before entering the internal steam pipe on its way to the stop valve, rendered the steam very dry and gave great security from priming even under rapid variations of power. Another advantage of the over-water tube is that in the event of a tube bursting, much less damage is done, the rush of steam keeping back the water.

When the boiler is not in use many engineers prefer to fill



, 46·8 sq. ft. Weight of boiler dry, with mountings, 9 tons.

the boilers completely with pure water to prevent corrosion (see Art. 8, Chap. XI.). In order to do this it is obvious that the highest point of every tube must be at its entry into the steam drum, since otherwise an air pocket would be formed. In order to prevent these air pockets modern Thornycroft boilers have been designed with some of their tubes entering the steam drum below the water line.

Plate V. shows the modified "Speedy" type designed for this purpose, and in Fig. 77 is illustrated the Thornycroft-Schulz boiler, which is a modification of the "Daring" type in which the arrangement of tube walls gives a somewhat different course to the gases, as shown by the arrows.

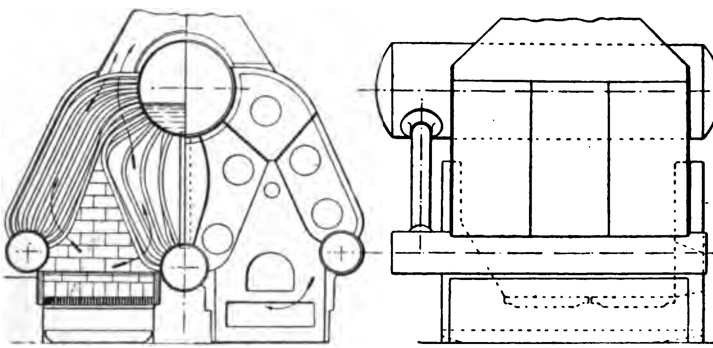


FIG. 77.—Thornycroft Schulz boiler (Thornycroft form).

18. Yarrow Boiler.—This boiler consists of an upper steam and water drum connected by several rows of straight tubes with two water drums or pockets as shown in Fig. 78. In the first design adopted, the steam drum and the water pockets were both made with bolted joints. Later on, when the boilers were made in larger sizes, the bolted joints in the steam drum were dispensed with, the joints being made of the usual riveted type, as shown in Fig. 78. The bottom ends of the tubes are expanded into a tube plate, to the under side of which the water pocket is bolted.

Fig. 79 shows the Yarrow boiler as fitted on H.M.S. *Triumph*. The weight of each boiler complete with water

was 34·5 tons and without water 29 tons, the boilers being constructed for a working pressure of 280 pounds per square inch. There were 1008 tubes in each boiler, $1\frac{3}{4}$ inch outside diameter and $\frac{5}{32}$ inch thick, except the two rows nearest the fire which were $\frac{3}{16}$ inch thick. The average length of the tubes

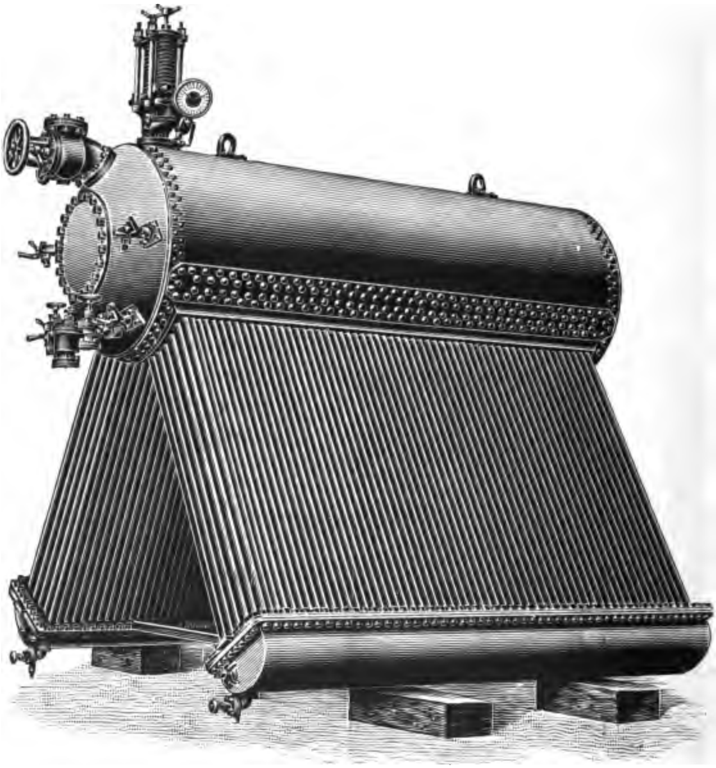


FIG. 78.—Yarrow water-tube boiler.

was 6 feet $9\frac{1}{4}$ inches, each tube being expanded into the tube plates as in Fig. 78. A row of distance pieces is fitted to each nest of tubes to maintain them at a uniform distance apart as shown. The feed water is introduced into the steam drum, the water circulation being down the outside tubes to the

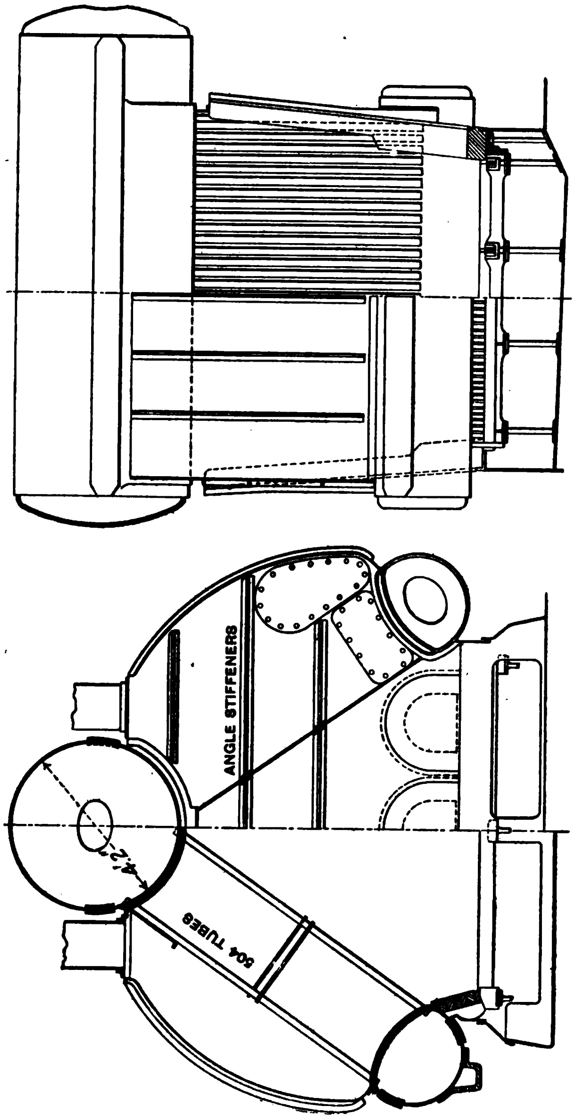


FIG. 79.—Yarrow boiler for H.M.S. *Triumph.*

two water drums and up the other tubes back to the steam drum in which the steam is liberated. The mud or sediment from the water collects in the bottom water drums, from which it is periodically blown out. One great advantage of the Yarrow boiler lies in the straight tubes, which are very accessible for cleaning and inspection.

No difficulty has been experienced in practice in keeping the tube plate joints steam tight. Although the tubes are straight, any small difference in length seems to be met by the elasticity of the material of which the tubes are made. Solid drawn steel tubes from 1 inch to $1\frac{1}{2}$ inch diameter are used, having an average thickness of about 0·08 inch. A casing of asbestos and steel envelops the boiler, the ends of the combustion chamber being closed by firebrick attached to sloping plates as shown in Fig. 79.

In the latest design, the water pockets are riveted up as well as the steam drum, and the feed water is admitted, not into the steam drum, but into the water pocket (see Fig. 80). The last two or three rows of tubes most remote from the fire are partitioned off by a longitudinal plate which acts as a feed collector, into which the feed water is admitted. The water is thereby compelled to ascend the tubes furthest from the fire; this not only increases the economy since the cold water takes more heat out of the gases at this place than would otherwise be the case, but also any grease or sediment that comes over with the feed water is deposited in these rows of tubes rather than in those nearer the fire. In conjunction however with this arrangement it is necessary to place longitudinal plates in the steam drum (Fig. 80) so as to prevent the cold water being instantly drawn down the adjacent tubes, the longitudinal plate making the feed water disperse generally in the steam drum.

19. The Yarrow Boiler fitted with Superheater.—In the ordinary type of Yarrow boiler, previously described, the temperature of the gases leaving the tubes is too low for superheating purposes; special arrangements have therefore to be made in order to prevent so much of the heat being abstracted by the ordinary boiler tubes. Fig. 80 shows the arrangement

recently described by Mr. Harold E. Yarrow.* The superheater is placed on the left-hand side of the boiler, above a lesser number of rows of generator tubes than are fitted on the right-hand side in order to maintain, as far as possible, equal resistance to the flow of gases on each side of the boiler. The total heating-surface of the boiler, experiments on which are described below, was 6700 square feet, of which 1265 square feet consisted of superheater surface; the total heating-surface on the super-

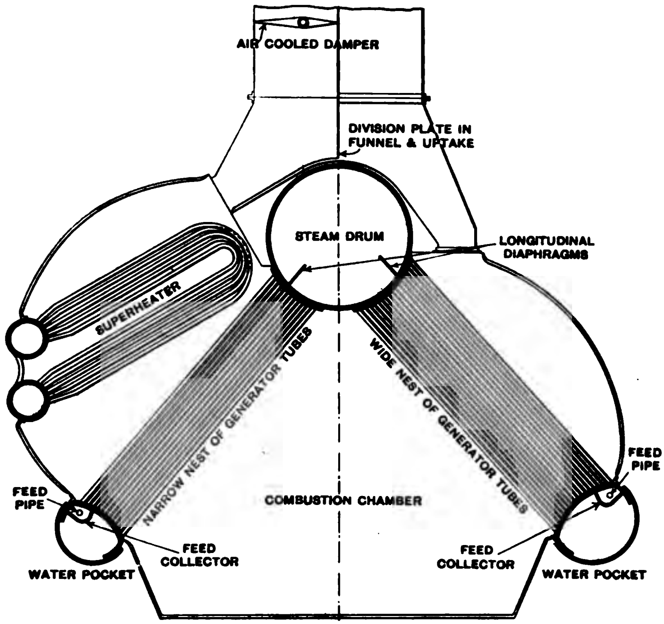


FIG. 80.

heater side of the boiler was 3453 square feet, and on the other side 3247 square feet.

The superheater consists of a number of U-tubes expanded into two longitudinal collectors, a damper being fitted in the

* Fig. 80 and 81 are taken by kind permission from Mr. Harold E. Yarrow's paper on "Superheaters in Marine Boilers," read before the Institution of Naval Architects, March 28, 1912.

uptake on the same side as shown in Fig. 80. The advantages claimed for this arrangement are twofold, namely:—

(1) On closing this damper, the whole of the gases are deflected towards the opposite side of the boiler and no hot gases pass through the superheater, hence, when raising steam, or if the main engines should be suddenly eased or stopped, the superheater is shut off and damage to the tubes is prevented (see Art. 20, Chap. VII.); in addition, the steam is not superheated to an excessive extent although there is only a small circulation of steam through the tubes.

(2) Closing the damper also greatly reduces the output of the boiler at the time when a reduced supply of steam is wanted, because only about one-half of the heating surface comes into contact with the hot gases.

To avoid distortion of the damper through overheating, it is fitted with a hollow spindle through which air circulates on its way to the space between the two plates of the damper.

Numerous experiments were carried out on this boiler, which was arranged for burning *oil* fuel. Pyrometers were fitted in various positions between the tubes and the temperature of the gases measured. Fig. 81 shows the temperature of the gases at various points of the boiler. The vertical lines correspond to the position of the pyrometers as shown in the lower part of the diagram. The upper curve indicates the gas temperatures at a rate of evaporation of 16 pounds of water per square foot of heating surface per hour, and the lower curve represents the gas temperatures at a rate of evaporation of slightly over 3 pounds per square foot of heating surface. The line BC represents the temperature of the steam at 200 pounds per square inch, and the line EF the temperature of the air-pump discharge, taken at 78° F.

Referring to Fig. 81, it will be seen that a large amount of heat is abstracted from the gases by the first few rows of tubes, as is evident from the great drop in temperature over this region. There is also a sudden drop in temperature where the gases pass through the last two rows of tubes at A and A′. This is due to the fact that the cold feed water (which enters a portion of the water pocket) abstracts a greater amount of heat

from the gases in ascending the two outside rows of tubes than would be the case if these tubes were full of water at the same temperature as the steam.

Table V. shows the results obtained with the damper

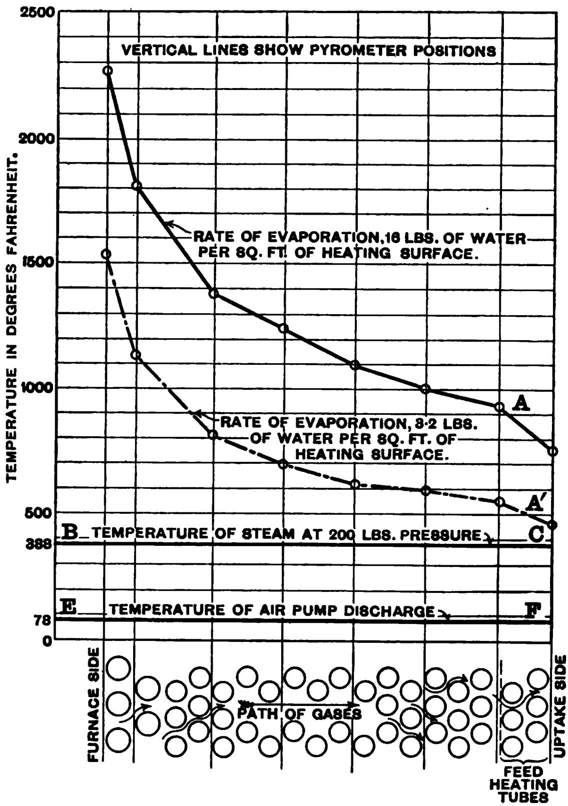


FIG. 81.—Temperature of the gases at various points in a Yarrow boiler.

open and the superheater working, and Table VI. the results of trials with the damper shut. It will be noticed that nearly 2 pounds of oil were burnt per square foot of heating surface per hour at full load with the damper shut, and Mr. Yarrow

TABLE V.—TRIALS WITH DAMPER OPEN.

Steam pressure, pounds per square inch.	Superheat in deg. Fahr.	Air pressure in inches of water.	Pounds of water evaporated per hour.	Pounds of oil fuel burnt per hour.	From and at 212 deg. Fahr.		Pounds of oil fuel burnt per square foot of heating surface per hour.	Temperature of feed-water in deg. Fahr.	Temperature between small nest of generator tubes and superheater in deg. Fahr.	Temperature of uptake in deg. Fahr.	
					Water per pound of oil.	Water per square foot of heating surface.				Above superheater.	Above large nest of generator tubes.
242·0	93·5	5·0	94,659	8286	14·6	18·0	1·237	58·0	1121	828	887
243·0	93·0	3·16	76,021	6454	15·0	14·4	0·9635	63·5	926	698	727
243·7	82·5	2·44	68,387	5695	15·2	12·9	0·850	63·5	903	685	688
242·8	61·1	1·7	46,041	3630	15·9	8·6	0·542	64·0	647	536	551
241·8	31·0	0·998	20,059	1540	16·1	3·7	0·230	62·2	481	432	448
242·2	20·75	0·625	8,478	649	16·1	1·55	0·096	63·5	465	409	416

Heating surface { Large nest of generator tubes . . . . . . 3247 sq. ft.; Small nest of generator tubes . . . . . . 2188 „; Superheater . . . . 1265 „ } 6700 sq. ft. total.

On these trials the heating surface is taken as the total heating surface of 6700 sq. ft.

TABLE VI.—TRIALS WITH DAMPER SHUT.

Steam pressure, pounds per square inch.	Air pressure, inches of water.	Pounds of water evaporated per hour.	Pounds of oil-fuel burnt per hour.	From and at 212 deg. Fahr.		Pounds of oil-fuel burnt per square foot of heating surface per hour.	Temperature of feed-water in deg. Fahr.	Temperature of uptakes, in deg. Fahr., above large nest of generator tubes.
				Water per pound of oil.	Water per square foot of heating surface.			
242·0	4·85	68,648	6287	13·25	25·66	1·936	61·0	913
242·25	3·97	57,693	5065	13·84	21·6	1·56	60·0	843
242·4	2·491	44,050	3504	15·3	16·5	1·09	60·3	673
242·5	1·46	31,481	2473	15·4	11·75	0·76	63·5	603

On these trials the heating surface of boiler is taken as heating surface of large nest of generator tubes = 3247 sq. ft.

states, as the result of these and other experiments, that with a boiler of this type it is quite possible to burn this amount of oil without injury to the boiler.

20. Hopwood's Vertical Water-Tube Boiler.—This boiler, made by Messrs. Marshall, Sons & Co., is shown in Fig. 82. It will be noticed that two of the sides of the firebox are flattened so as to form tube plates. A number of inclined water tubes are carried across the upper part of the firebox and fixed in these tube plates, the tubes lying a little higher at one end than at the other. These tubes are placed directly over the fire and constitute a large proportion of heating surface of the boiler. The ends of the tubes are opposite two large manholes which have strong steel external covers with faced joints as shown, so that the tubes can be easily cleaned or removed when required. The flat top of the firebox is stayed to the crown of the boiler by a number of longitudinal stays, and the boiler is designed for a working pressure of 100 pounds per square inch. The boiler is simple in construction and has one great advantage over the multitubular or smoke-tube type already described (Art. 11, Chap. VI.), namely, that it is much easier to clean the *inside* of the water tubes than the *outside* of the smoke tubes in the latter type of boiler.

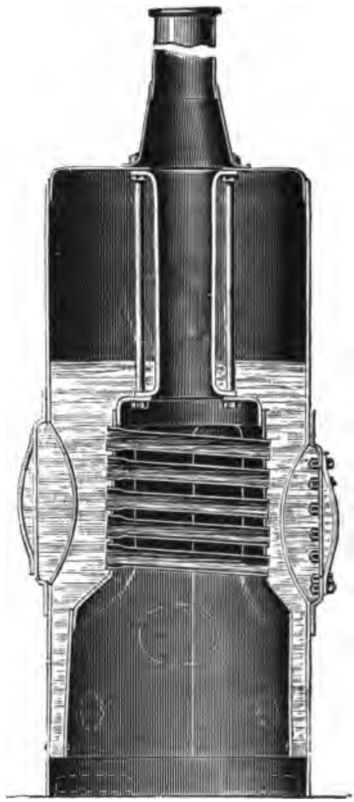


FIG. 82.—Hopwood's vertical water-tube boiler.

CHAPTER VIII

BOILER ACCESSORIES

1. Feed Pumps.—For supplying feed water to a boiler, both pumps and injectors are extensively used, but it is impossible to lay down any hard and fast rules to decide in all cases which is the best to adopt. As far as economy alone is concerned an injector is better than a pump because, neglecting radiation losses, all the heat contained in the steam which works the injector is returned to the boiler in the feed water and in addition to performing the work of a pump, the injector acts as a feed water heater. Economy alone, however, is not the only thing to be considered, reliability in working being of equal importance. The usual practice on board ship and also in large land installations is to use feed pumps, because of their greater reliability and the lesser amount of attention which they require when a large amount of feed water is necessary. On the other hand, when a single boiler of comparatively small size is used there is little to choose between a pump or an injector, the latter being perhaps the more convenient although opinions may differ on this point. Injectors are almost always used on locomotives in preference to feed pumps.

The great variety of feed pumps used in practice may be broadly classified into two types, namely, those driven by a belt or other means, the ram of the pump receiving its reciprocating motion through a connecting rod driven by a crank; and those having no rotary motion at all, this type being known as the direct-acting pump.

In high-speed pumps, or in cases where the delivery pipe is very long, the accelerating force required on the pump plunger, and therefore on the water, causes a large increase in

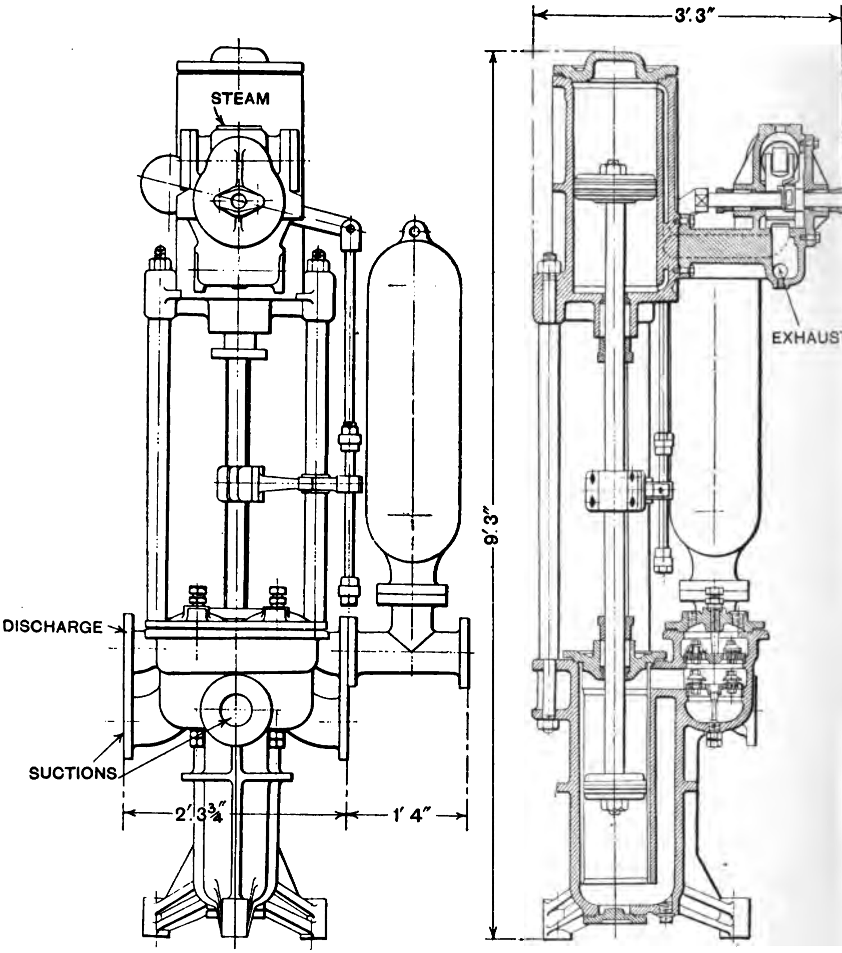
the stresses in the different parts of the pump; it may also under certain conditions lead to separation of water in the pipe, and violent hammer action with all its attendant evils may be set up. To reduce the effect of these inertia forces, air vessels are usually put on the delivery pipe. For the same reasons, if the pump runs at a high speed and the suction pipe is long, an air vessel will also be required on the suction side. For simplicity in construction and smoothness in running a slow-speed pump is desirable on account of the very much smaller inertia forces.

Fig. 83 shows a section through a vertical slow-speed direct-acting pump made by Messrs. Clarke, Chapman & Co. The suction and delivery valves of the pump consist of a number of small valves arranged on a circular seat, giving thereby a large valve area with a small lift. The valve on the upper steam cylinder is of the Corliss type and is arranged to cut off before the end of the stroke, thereby reducing the speed of the piston and allowing the water valves on the pump cylinder to close quietly without shock.

Fig. 84 is a sectional view showing the position of the steam valves just before the piston valve G is actuated by the steam, while Fig. 85 shows the position of the valves when the piston valve G has been actuated by the steam and main distributing valve C, which gives steam and exhaust to the cylinder through the ports T and B respectively.

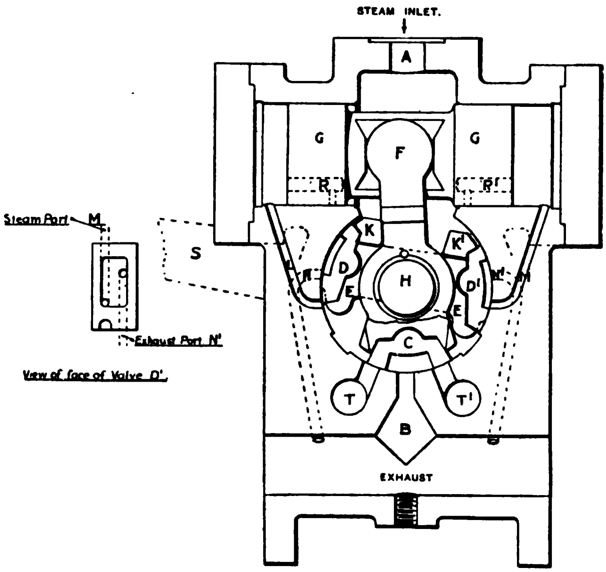
The main distributing valve C, and the auxiliary or pilot valves D and D′, are merely slide valves which are rotated on a cylindrical face. The two auxiliary valves are held in position by the carrier E, which has two lugs K and K′ on the top portion of it, and the main distributing valve is held by the arm F, all three being loose pieces kept up against the face of the valve chest by the pressure of the steam on the back. The carrier E is a forging, having the spindle H extended on both ends, upon which the arm F rotates. The spindle H has attached to it an outside lever S fitted with a tappet rod which is worked by a crosshead on the pump rod. G is a small piston which is moved backwards and forwards—in part mechanically, the motion being completed by the steam pressure—and engages

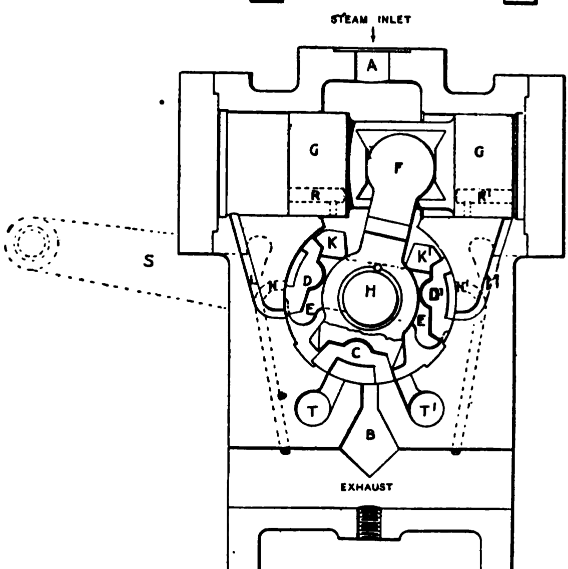
the arm F, which carries with it the main distributing valve C.



Woodeson's patent steam cylinder, 10½ ins. dia. by 21 ins. stroke; water cylinder, 8 ins. dia.

FIG. 83.—Clarke-Chapman direct-acting pump.





FIGS. 84, 85.—Woodeson's patent Corliss type valve gear.

When the main piston is nearing the end of its stroke, and the crosshead on the pump rod engages the valve gear by means of the external lever S, it brings one of the projections K or K′ of the carrier E into contact with the arm F. It thus moves the whole valve gear, including the piston G, the main distributing valve C, and the two auxiliary valves D and D′, mechanically, until the ports L and M are open to either steam or exhaust; the movement of the piston G is then completed by means of the steam, the piston G carrying with it the arm F and also the distributing valve C.

It will be seen that there is a certain amount of clearance between the arm F and the lugs K and K′. This enables the piston G to be put in equilibrium before the carrier E comes into contact with the arm F on the return stroke of the main piston, by allowing the auxiliary valve D or D′ to close the port L or M, as the case may be, from the exhaust, and open it to steam so that the piston G has steam at both ends and is thereby in equilibrium, all this being done before the lug K or K′ comes into contact with the arm F. By this means the work on the outside lever S is reduced to a minimum.

The two passages R and R′ are to allow for displacement of steam during the time that the piston G is being mechanically moved, and the arrangement is such that as soon as either of these passages are closed by the piston G, and compression of steam begins, then the auxiliary value D or D′ is in the correct position to immediately relieve the compression by opening the closed end to the exhaust N or N′, and the remainder of the travel of G is thus completed by the action of the steam.

An important feature of this valve gear is that, should the steam fail to complete the movement of the piston G, then the mechanical movement is continued until the main steam port is opened to the opposite end of the main cylinder, so forming a steam cushion to bring the piston to rest.

2. Injectors.—Injectors may be divided into two classes, "Non-automatic" and "Automatic" or Self-acting Injectors.

Non-automatic Injectors do not restart automatically if for any reason the discharge is interrupted, and further, both the steam and water must be independently regulated by hand

when starting in the first instance, or when restarting in the event of any discontinuity in the discharge.

Non-automatic Injectors are further divided into two classes, namely:

(*a*) *Non-Automatic Lifting Injectors*, which lift their feed water and usually have adjustable cones, *i.e.* their cones can be moved backwards or forwards, which increases or reduces the area of both suction and discharge so as to feed against a lower or higher steam pressure. A larger area also suits a higher temperature of the suction supply of water.

(*b*) *Non-Automatic Non-Lifting Injectors*, which do not lift, but require their feed water to flow to them. They have fixed cones and are adjusted for varying pressures by a water cock.

Automatic or Self-Acting Injectors, on the other hand, lift their feed water if required and start working as soon as steam and water are turned on, without any manipulation. They will also restart instantaneously and automatically should the jet be accidentally broken.

3. Giffard's Injector.—A section of this injector, which is of the Non-Automatic Lifting type, is shown in Fig. 86. Steam from the boiler is admitted through the opening shown and passes through a conical nozzle to the injector. The quantity of steam admitted is regulated by the opening of this nozzle. The lower end of the vertical spindle is turned conical in order to fit the nozzle, and by means of the hand wheel at the top it may be screwed up or down so as to increase or decrease, as the case may be, the amount of steam admitted. The boiler feed water enters the injector a little below the steam inlet on the opposite side as shown, and passes round the outside of the steam nozzle, the supply being regulated by means of the hand wheel at the side. This hand wheel moves the internal conical tube up or down by means of the rack and pinion shown, thereby regulating the area of the annular passage through which the water passes. The branch pipe below the water inlet is for the water overflow, and the bottom of the injector, which is fitted with a non-return valve, is connected by piping to the feed check valve on the boiler.

The following simple explanation will enable the reader to understand the principle on which the injector works.

Some of the original heat energy in the steam is converted into kinetic energy as the steam rushes through the steam nozzle, and when moving at a very high speed the steam is condensed on coming into contact with the surrounding feed

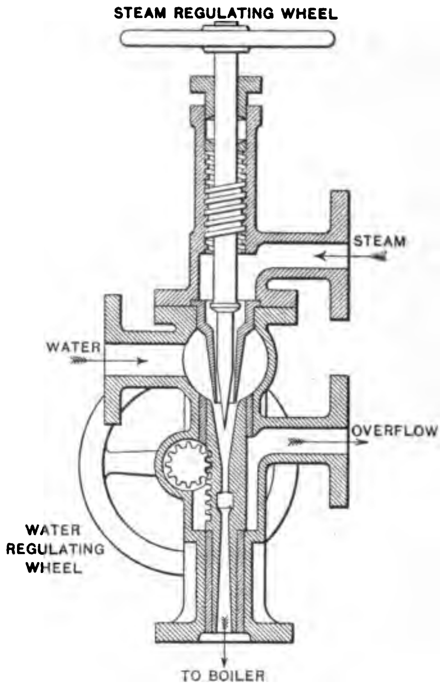


FIG. 86.—Giffard's injector.

water, but its speed is not materially reduced. The partial vacuum formed in the injector by the condensation of the steam results in more feed water rushing in to be carried by the condensed steam jet into the boiler; the velocity of the steam jet is very much higher than the velocity with which a jet of water would be issuing through the same nozzle under the same

pressure, and it is this fact which enables the water to enter the boiler against the pressure of steam inside it.

4. Holden & Brooke's Automatic Injector.—With this injector one movement of the lever A (Fig. 87) performs the following operations simultaneously:—

1. Turns steam on and starts the injector.
2. Regulates the injector to suit any required pressure.
3. Adjusts the quantity of feed as required.

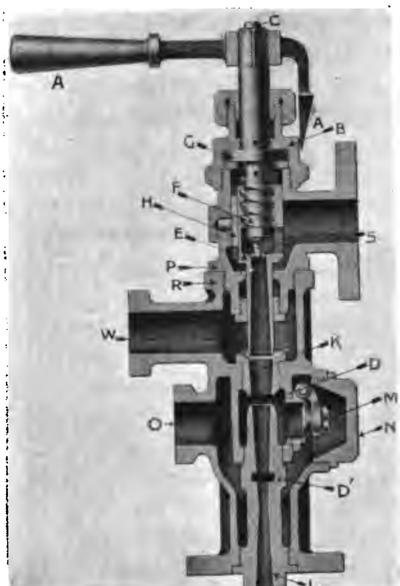


Fig. 87.—Holden & Brooke's automatic injector.

When the pointer on the lever A is set to the position marked "shut" on the regulating scale B, which is graduated for various steam pressures, the steam cone E is raised until, coming against the coned end of the spindle F, the passage through the steam nozzle is closed and the steam consequently shut off.

On starting the injector by moving the lever and pointer A, the spindle F revolves, and in doing so forces the steam cone E away from it, and so opens the steam passage. When this passage is just open the steam supply is at its minimum, and the water supply, which is regulated by the distance between the end of the steam nozzle E and the water cone K, is at its maximum. In such a position the injector is suitably set for the highest steam pressures.

As the steam nozzle E moves further away from the spindle F, the steam space is increased and the water space is at the same time correspondingly decreased. The angles at the end

of the steam cone and at the opening of the water cone are so arranged that the supply of steam and water (dependent upon the position of the steam cone) is always regulated to suit any pressure on the scale B, to which the pointer A may be set. The spindle F is prevented from moving longitudinally by the collar G, and the steam cone is prevented from rotating by means of the feather key H. The turning of the spindle, therefore, causes a longitudinal motion only of the steam cone.

On starting the injector the first rush of steam escapes through the overflow valve M and out through the overflow passage O, but immediately the jet is formed the valve M closes, and is kept tight down on its seat by the vacuum due to the jet as it rushes past the gap D. Any further overflow from the jet takes place at the overflow gap D, and also escapes at O, such overflow being checked by moving the pointer A to the proper point. The only portion of the injector at all liable to incrustation is the delivery cone L and the lower combination cone attached to it, and these can be readily withdrawn for examination or cleaning by unscrewing from the body.

The regulation of the feed may be effected while the injector is working by moving the pointer A slightly to the left or to the right of its working position. Decreasing the quantity of feed increases its temperature, and the delivery can by this means be reduced to about half of the maximum.

Fig. 88 shows the general arrangement of the injector feeding a Lancashire boiler, and Fig. 89 the arrangement when fitted to a Babcock and Wilcox boiler. The injector, being automatic and adjustable, works either lifting or non-lifting. In Fig. 88 the dotted lines show the water connection for working non-lifting. A feed check valve on the boiler is shown, and a strainer on the end of the suction pipe; the water cock is only used when the injector is non-lifting.

The arrangement shown in Fig. 89 is shown fitted with a branch B, to which a feed pump may be coupled for use when the injector is not working. A is the boiler feed check valve; C the feed stop valve, which shuts off the injector when the pump is running; D the injector, and E the steam stop valve for the injector. The greater height of the boiler makes it advisable to

have the extra valves C and E, in order that the injector may be conveniently controlled, all the necessary connections being within easy reach of the attendant.

5. Exhaust Steam Injectors.—Exhaust steam injectors form the most economical method of supplying feed water to a boiler, since the heat in the exhaust steam from the engines is

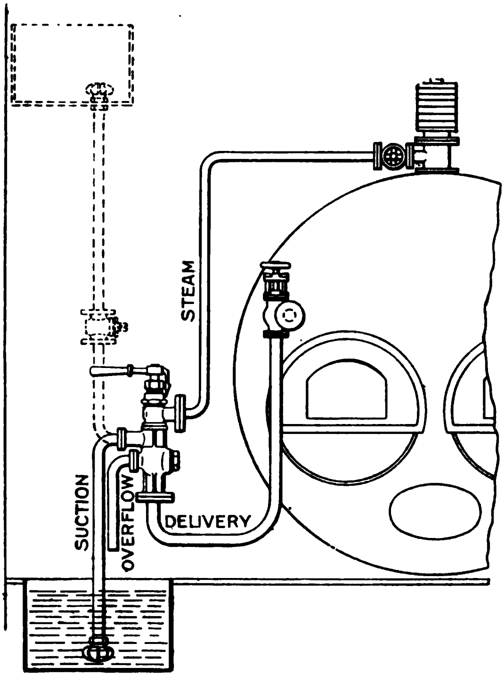


FIG. 88.—Injector fitted to Lancashire boiler.

returned to the boiler in the feed water. Their use results in either the same amount of steam being generated with less fuel, or in much more steam with the same amount of fuel. They can be attached to any form of non-condensing engine, and are usually of the non-lifting, self-acting or re-starting type. One type of Holden and Brooke's exhaust steam injector is shown

in Fig. 90, which is self-explanatory. The type illustrated works entirely with exhaust steam at about atmospheric pres-

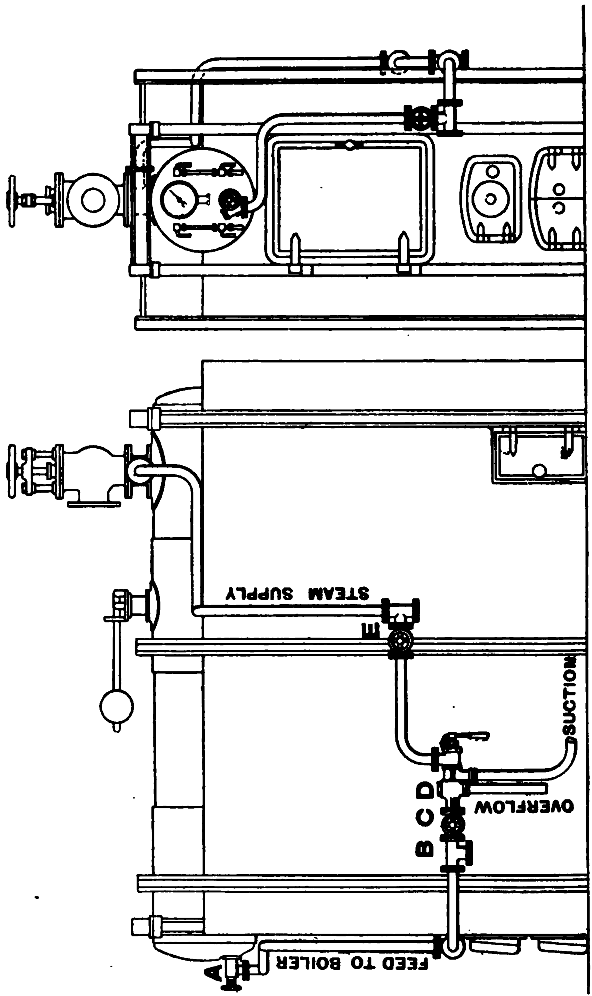


Fig. 89.—Injector fitted to Babcock & Wilcox boiler.

sure, and is made to feed against boiler pressures of from 65 to 95 pounds per square inch, according to size, delivering the

feed water at temperatures up to 190° F. For higher boiler pressures, a small jet of live steam is added by attaching a live steam connection to a tapped boss provided on the wing valve, as shown dotted at B. The addition of the live steam jet does not diminish the quantity of exhaust steam used by the injector, and as it increases the already high temperature of the delivery, the beneficial effects on the economy and capacity of the boiler are further increased. The live steam connection also enables the boiler to be fed when the engine is not at work. In all ordinary types of exhaust steam injectors the temperature of the water supply should not exceed about 75° F.

For working with water at a higher temperature than 75° F. automatic hot-water injectors are now made, which, when fixed as non-lifting, will feed against a boiler pressure of 200 pounds per square inch and take water at 130° F.

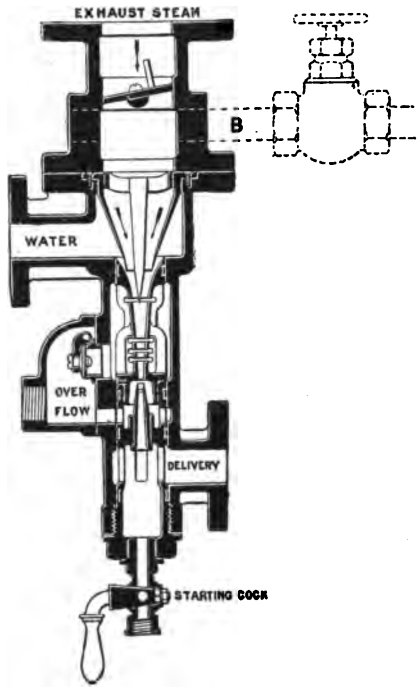


FIG. 90.—Holden & Brooke's exhaust steam injector.

6. Exhaust Steam Injector applied to Locomotives.—As already mentioned (Art. 1), injectors are almost always used on locomotives in preference to feed pumps. Messrs. Davies & Metcalfe, Ltd. have given special attention to the application of exhaust steam injectors for feeding locomotive boilers. The general arrangement adopted by this firm will be understood by referring to Fig. 91. The injector

consists of two parts known as the exhaust portion and the supplementary portion, these two portions being connected by

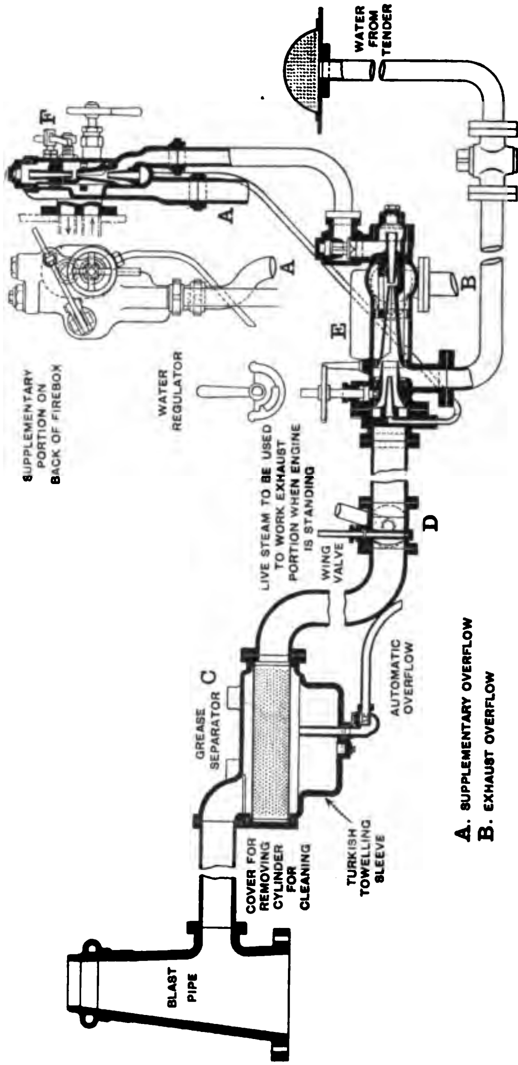


FIG. 91.—Arrangement of exhaust steam injector on a locomotive (Messrs. Davies & Metcalfe, Ltd.).

means of a pipe and the exhaust portion delivering into the supplementary portion.

Exhaust steam is taken from the blast pipe to the grease separator C. The function of this part is to remove from the exhaust steam all particles of grease, dirt, or condensed steam, which may have been carried along by the exhaust steam. The steam on entering the casing strikes against the baffle plates, which deflect the suspended matter into the well below, from which it is discharged into a drain pipe through an automatic drip valve. The steam then passes through a perforated metal cylinder, covered by a sleeve made of absorbent towelling, which further dries it and removes any impurities remaining in the steam before it passes to the throttle valve D. This is of the ordinary wing valve type, and is worked from the foot-plate of the engine; it is used to shut off the exhaust steam from the exhaust portion when the injector is not required to feed the boiler.

There is also coupled to the throttle valve casing a branch pipe, which conducts live steam to the injector, for use in working the exhaust portion when the regulator is shut and no exhaust steam is available.

From the throttle valve D the steam is led to the exhaust portion E, which is a complete exhaust steam injector in itself, consisting of the usual three nozzles or cones, viz.: the steam, water, and delivery nozzles; it has steam, feed water, overflow, and delivery connections. The exhaust steam nozzle is of a very large bore, and a nozzle of very small diameter, known as the "Inducer" nozzle, supplied with live steam, is fixed concentrically with it, its object being to create a vacuum in the exhaust nozzle, and so to induce a greater flow of exhaust steam from the blast pipe into the injector.

The water regulation is effected by moving the exhaust steam nozzle backwards or forwards. In this way, the surrounding area between it and the mouth of the combining cone is varied, and consequently the volume of water flowing into the combining nozzle is regulated according to the quantity of water required. The movement is effected by an eccentric pin, which is worked from the foot-plate. The delivery from the

exhaust portion E is connected by a pipe to the feed of the supplementary portion F, and it is essential that the exhaust portion must be fixed *below* the level of the water in the feed tank, since in common with most exhaust steam injectors it is of the non-lifting type.

The supplementary portion F is simply a small live steam injector, specially designed for handling hot water. It is capable of working up to a pressure of over 300 pounds per square inch, receiving feed water at from 170° F. to 190° F., from the exhaust portion, and delivering its water to the boiler at a temperature of about 280° F.

In the design of the supplementary portion, a serious problem was encountered, in the necessity for a weighted overflow valve, which is rendered necessary by the high temperature of the water delivered to the boiler. Since the temperature is above 212° F. (the boiling point of water at atmospheric pressure), the overflow valve must be weighted for a pressure higher than that corresponding to the temperature of the delivery water, otherwise the water would be converted into steam and discharge through the overflow. At the same time, the overflow valve A must act automatically, so that if the continuity of the jet be broken in any way (as by jars over points, sudden shocks, etc.), the overflow valve must open and allow a free discharge until the injector restarts itself and restores the continuity of the jet. This automatic action is accomplished immediately and with practically no waste of water or steam by the valve A. A lever is pivoted on a fulcrum on the injector casing, one end bearing on the top of the overflow valve, the other end against the top of a small piston which is in communication with the delivery of the supplementary portion.

When steam is turned on in the supplementary portion, and until steady working is obtained, the pressure under the overflow valve keeps it open and allows a free discharge into the overflow pipe. Directly the continuity of the jet is established, the pressure in the delivery, acting under the small piston, raises it, and so closes the overflow valve and keeps it tight on its seating. If the jet breaks from any cause, the pressure on

the piston is reduced, the overflow valve opens, and remains open until the injector restarts.

7. **Feed-Water Filters.**—The question of filtering the feed water is one of great importance, particularly in the case of small-tube water-tube boilers. The impurities present consist either of particles of mud or oil held in mechanical suspension, or of certain salts chemically dissolved in the water. In the former case the suspended matter may be removed by filtration, but in the latter case recourse must be had to chemical precipitation (see Chap. X.). When the quantity of mechanically suspended matter (other than grease) is small, special treatment may be unnecessary; but when there is a large amount, it is advisable to filter the water before it is used.

In condensing plants the exhaust steam from the engines is condensed and used as hot feed water for the boilers. Such water will invariably contain a certain amount of oil or grease from the engine cylinders which if allowed to enter the boiler is very detrimental. When superheated steam is used, special arrangements are made to ensure efficient lubrication of the piston, piston rod and valve faces, with the result that in many cases more oil is present in the exhaust than if the engines were supplied with saturated steam. The complete removal of oil from water is a matter of great difficulty and direct filtration cannot always be relied upon for this purpose. On account of the difficulty of removing all the oil from the feed water it is always best to fit oil separators in the exhaust from the engines, because it is found much easier to remove the oil from the steam than from the water after condensation. When an exhaust steam injector is used, an oil separator is invariably fitted in the exhaust pipe to prevent any oil in the steam from finding its way with the feed water into the boiler.

Many designs of both water and oil filters are used in practice. Figs. 92 and 93 show a pressure water filter made by Messrs. Mather & Platt. The filter consists of a closed cylinder having parallel sides, with dished ends at the top and bottom, constructed of riveted steel plates. The filtering medium A, consisting of graded quartz crystals, rests on a false bottom B, riveted to the shell of the filter and fitted with specially

shaped nozzles (not shown) screwed in from the underside; ready access to the nozzles can be obtained by means of a manhole fitted in the side of the collecting chamber D, and they can be inspected and replaced, when necessary, without

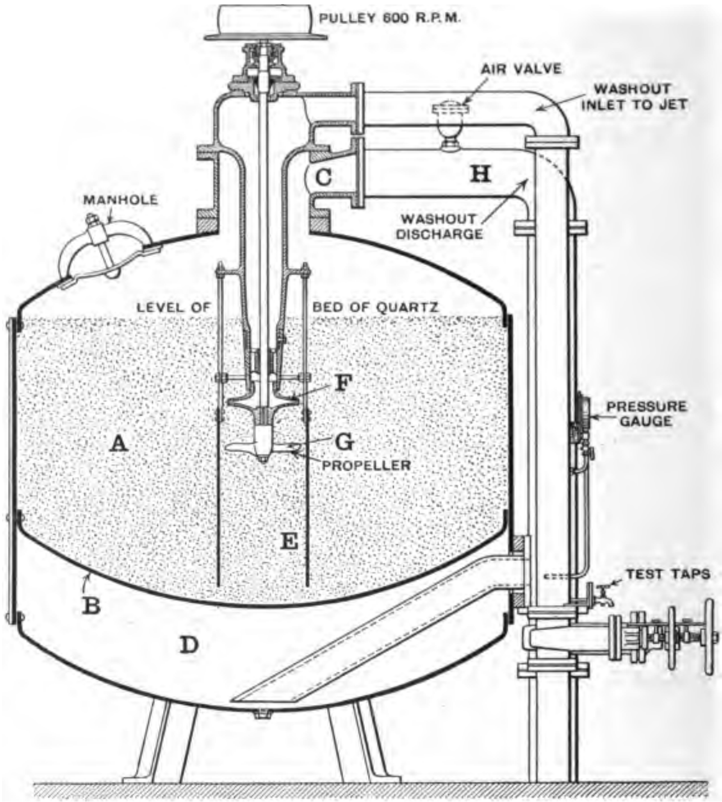


FIG. 92.—Mather & Platt's water filter (section).

in any way disturbing the quartz bed. By this arrangement, each nozzle is kept as a distinct unit which can be removed and replaced in a few minutes.

The object of the nozzles is to ensure the effective use of all parts of the filtering bed, as well as the proper distribution of

the water used for washing. The nozzles are screwed direct into the false bottom B, and the filtrate as well as the wash water passes through them.

The unfiltered water enters at the top of the filter through the opening at C and is evenly distributed over the bed. After

FIG. 93.—Mather & Platt's water filter.

percolating through the bed it flows into the collecting chamber and leaves by the filtrate valve.

For cleansing the filtering material a central vertical tube E is fixed in the middle of the bed A, extending upwards to within a few inches of the top level of the filtering material. Suspended

in this tube is a steel shaft on which are fixed at a short distance above the top of the tube, two jets F which are connected to the filtrate main, and blades G in the form of a propeller situated in the upper end of the tube; the shaft is driven from outside the filter by means of belting, gearing, electric or other motor, at about 600 revolutions per minute.

When it is required to wash the bed, filtered water is admitted to the underside of the false bottom, and rising up through the nozzles puts the bed in suspension. The shaft with the jets and propeller thereon, is then caused to revolve at a high speed, and the quartz is thoroughly cleansed by being drawn up the vertical tube E and passed over the top, where it comes in contact with clean water issuing from the jets at a high velocity. The filtering medium then sinks in the filter, and the floating impurities are carried away with the wash water through the washout discharge pipe H at the top of the filter.

The filter is fitted with a pressure gauge connected to the inlet, filtrate and jets, and the exact pressure on any one of these pipes can readily be ascertained. Draw-off taps are fitted to the inlet and filtrate pipes to enable samples to be taken.

8. Oil Filters.—The Mills-Berryman "Sentry" filter made by Messrs. Isaac Storey & Sons, is shown in Fig. 94, and consists of an outer casing A in which a perforated basket B is enclosed. This basket is filled with some filtering material, usually woody fibre, through which the water passes first and in which the bulk of the oil and any solid suspended matter is retained, after which the water passes through one or two thicknesses of canvas, usually with a velocity of about 6 inches per minute. The perforated basket B can be readily removed and another one substituted ready packed for use. The filter should always be fitted on the inlet side with a pressure gauge whose reading, when too high, will show when the filter requires cleaning. A bye-pass valve should also be fitted in order that the feed may flow uninterruptedly in cases of emergency without passing through the filter.

9. Steam and Oil Separator.—Messrs. Holden &

Brooke's "Parallel Flow" oil separator is shown in Fig. 95. In this system of oil separation, the steam is led between the walls of a number of vertical passages in parallel, the total area of these passages being sufficient for the passage, *without back pressure*, of the total volume of steam for which the apparatus is designed. These walls form the depositing surfaces, and are provided at frequent intervals with vertical channels to catch and lead the water and grease downwards to the collecting chamber; they are continuous, and consequently, keep the steam under a continuous cleansing process. It will be seen

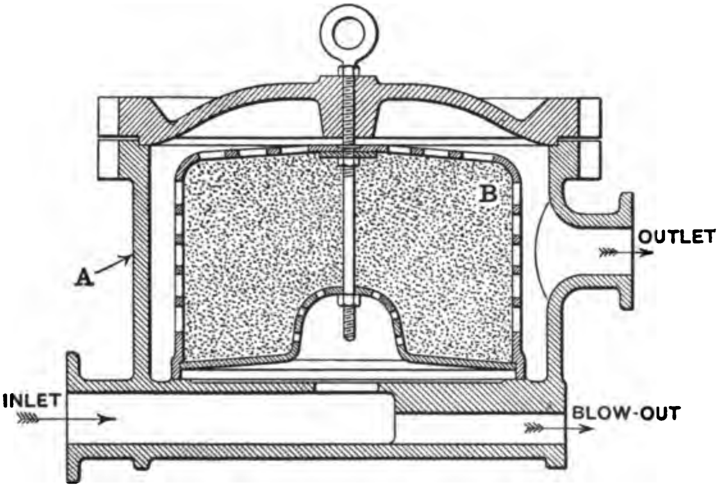


FIG. 94.—Mills-Berryman feed-water filter.

that the steam passes from the sides inwards to the centre, and thence outwards again twice during its passage through the separator. The total area of the several parallel passages through which the steam is made to flow is ample for its volume, while at the same time the closeness of the walls of these passages is such as to ensure good contact with the depositing surfaces. In Fig. 95, the walls of the main channels, through which the steam is caused to pass, are indicated by the thicker plates, and the subsidiary passages, by which the steam is broken up into a number of thin parallel

currents (so as to bring it and keep it in contact with the depositing surfaces) by the thinner plates. The channels which

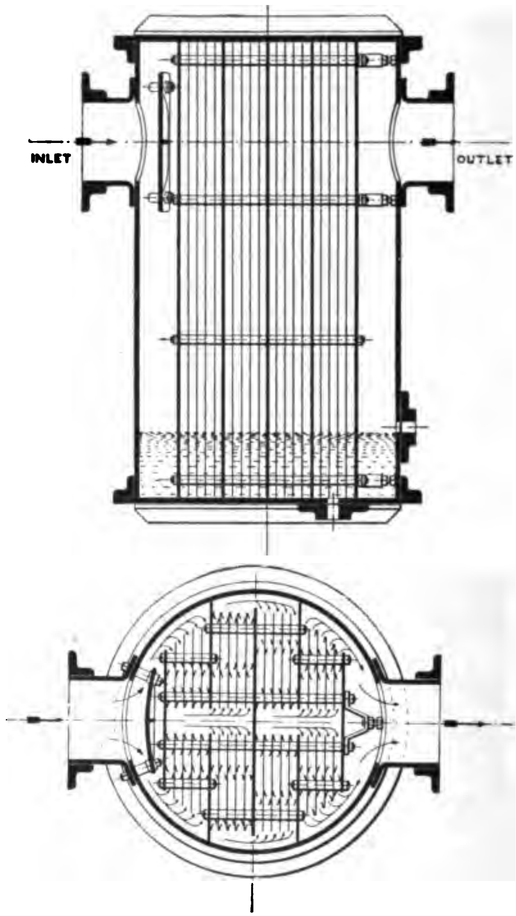


FIG. 95.—Holden & Brooke's oil separator.

are provided for leading off the oil and water, and which run vertically downwards the whole length of the plates, are shown on the plan view.

10. Feed-Water Heaters.—Considerable advantages are obtained in practice through heating the feed water before admission to the boiler. In the first place, the higher feed temperature results in a higher and more uniform mean temperature in the boiler; this lessens the differences of temperature in various parts of the boiler, and the stresses from expansion and contraction are correspondingly reduced. This is a matter of considerable importance in the Lancashire and similar types of boiler. Again, the quicker the water can be turned into steam, the more rapid will be the circulation, and the more rapid the circulation, the greater will be the heat transmitting power of the heating surface; hence with a hot feed a higher rate of evaporation is possible than with a cold one. The function of a boiler being to generate steam, it is obvious that the higher the feed temperature, the less will be the amount of heat required to raise the temperature of the water in the boiler to the temperature of evaporation; therefore, for the same fuel consumption more steam will be produced, again resulting in a higher rate of evaporation. In the ideal boiler, the temperature of the feed water would be the same as (or very little lower than) that of the steam, in which case the boiler would only have to supply the latent heat necessary for evaporation.

There are several methods of heating the feed water, all of which are extensively used in practice, namely:—

1. By means of the waste furnace gases leaving the boiler.
2. By means of exhaust steam from the engines.
3. By means of live steam.

The first two of the above methods result in a direct gain. On land the temperature of the flue gases leaving the boiler can be reduced by passing the gases through an economiser, and a saving is thereby made in the amount of heat carried away by the gases up the chimney. Feed heating by exhaust steam is more advantageous when non-condensing steam engines are used. In non-condensing engines the exhaust pressure is above atmospheric and the temperature of the exhaust steam is about 220° F.; hence the feed water can be heated up to about 180° F., whereas in condensing engines if the vacuum be about 26 ins., the temperature of the exhaust

steam is only 126° F., and the maximum temperature of the feed water must be somewhat below this.

Feed heating by means of live steam taken direct from the boiler is very frequently used, especially on board ship. Although no direct gain in economy results from this system (the heat in the steam taken from the boiler being merely returned to the boiler in the feed water as is the case when injectors are used), it is advantageous for the reasons already mentioned, *i.e.* lower temperature stresses owing to a more uniform temperature, more rapid circulation and increased rate of evaporation.

In Weir's system, the steam used for heating the feed water is taken from the receiver of the main engines and the exhaust from auxiliary engines. Some of this steam has already done work in the high-pressure cylinder of the main engine, and, neglecting radiation losses, all the heat it contains is returned to the boiler in the feed water. The resulting gain is not so great as with exhaust steam heaters, but it has the advantage of a higher feed temperature being obtained.

11. Green's Economiser.—This apparatus (Fig. 96) consists of a series of cast-iron pipes, 9 feet long, $4\frac{9}{16}$ inches external diameter and $3\frac{11}{16}$ inches internal diameter. The pipes are arranged in rows, each row containing four or more pipes depending on the size of the economiser; they are placed across the main flue between the boiler and the chimney. The sections containing each row of pipes are formed by forcing the pipes by hydraulic pressure into junction boxes or headers, the ends of the pipes being turned and the sockets in the junction boxes bored to receive them, thus forming a good metal to metal joint. When erected in position, the sections are connected by their top and bottom headers to multiple-flanged branch pipes. The bottom headers project through the front wall of the economiser chamber and are rendered accessible for cleaning purposes by means of access lids on the branch pipe opposite each header. The top headers are machined along the edges and form an air and gas tight roof.

The water is admitted to the economiser through the bottom branch pipe to the sections, and is collected in the top branch

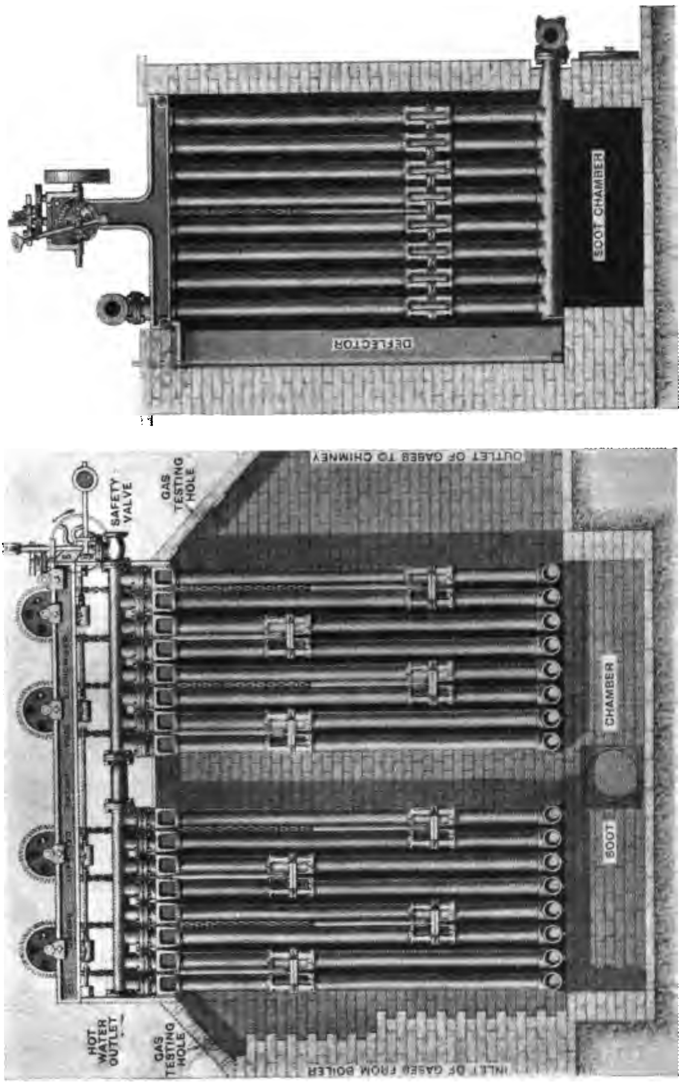


FIG. 96.—Green's economiser, 128 tubes.

pipe and delivered from it to the boiler feed-water pipe. Thermometer pockets are inserted at the inlet and outlet ends respectively of the bottom and top branch pipes. The flow of water and gases is in opposite directions, the feed water being introduced at the end nearest the chimney and taken out at the boiler end.

The exterior of the pipes is kept clean by scrapers attached to chains and actuated by overhead gearing which is driven by a small steam engine or electric motor. The power taken to drive the scrapers is found in practice to be about 0·75 horse-power for every 100 pipes.

The economiser is fitted with one or more safety valves according to its size; one or more blow-off valves are also fitted in such a manner as to drain the economiser completely. Water from the safety and blow-off valves (not shown in Fig. 96) should be discharged some distance away from the economiser, and not allowed to drain into the soot pit flues underneath, where its presence would involve the serious risk of corrosion of the pipes and bottom headers. The discharge should also be visible, in order that valve leakage and the condition of the water may be detected.

If the inlet temperature of the water to the economiser is low (40° F.), there is the risk of sweating or corrosion taking place on the outside of the pipes. By feeding with water at, say, 90° F. this difficulty is obviated, but when this is impossible, special provision is made on the economiser whereby the corrosion is located in two or three sections at the feed-water inlet end of the economiser, the remaining part being protected by the preliminary heating thus effected.

The size of an economiser should be regulated by the temperature of the gases leaving the boilers, the quantity of water evaporated, and the chimney draught. In the case of a Lancashire boiler 8 feet diameter and 30 feet long which would evaporate about 8000 pounds of water per hour at a pressure of, say, 160 pounds per square inch, it is usual to instal an economiser containing about 120 pipes. This should raise the temperature of the water which enters the economiser from the hot well (in a condensing plant) at a

temperature of about 90° F. to about 300° F. when delivered to the boilers.

When the boilers are working on natural chimney draught it is not advisable to cool the gases below 350° F.; otherwise the draught will be impaired (Art. 3, Chap. IV.). Although an economiser may be large enough to heat the water up to a high temperature, it depends entirely on how it is installed and worked if the most economical results are to be obtained from it. The flow of water through the economiser should be kept as uniform as possible, *i.e.* the water level in the boilers should be kept constant. Also there should be no air leakage. Very frequently the cause of a low temperature of the gases leaving a boiler is the result of unskilful firing, the grate not being kept properly covered with fuel, resulting in more cold air being supplied than is necessary for complete combustion (see Art. 6, Chap. III.). Openings in the brickwork around the fronts and backs of boilers, also badly fitting boiler dampers, all tend to reduce the temperature of the gases and increase the losses, with consequent loss of efficiency. It is not an unusual thing to find a substantial reduction of the temperature of the gases in their passage through an economiser, without a corresponding amount of heat being transmitted to the water in the pipes, this difference in temperature being often entirely due to the bad state of the brickwork of the flues and economiser.

In order to prevent air leakage through the brickwork, the walls enclosing the economiser should be built with hard-pressed bricks and cement. Glazed bricks are better still, and their extra cost would soon be recovered by the economy in fuel resulting from their use. If ordinary brickwork is used, it should be coated on the outside with pitch or other viscous substance to reduce porosity. The practice of encasing economisers with wrought-iron panelling, having 2 inches of asbestos yarn between the inner and outer plates, is sometimes followed; this, however, is an expensive remedy, but the heat saved, together with the greater facilities for access and repair, provides a reasonable return for the extra outlay.

Chimney draught should be controlled by the outlet damper

of the economiser, so that when the steam pressure is high, the closing of this damper keeps the gases among the economiser pipes. If the boiler dampers are used for this purpose, some of the heat contained in the flues, the economiser chamber, and in the feed water is abstracted, resulting in a reduction of both the draught and the efficiency of the economiser. In the case of a range of boilers of which several work at different pressures, it would be impossible to regulate the draught entirely by this damper, but where there is one boiler only, or several working at the same pressure, the outlet damper of the economiser should always be used. In addition to the increased efficiency obtained by this method of regulating the draught, it is very much easier for the fireman to manipulate one damper only than the six dampers in the side flues of three Lancashire boilers.

Radiation losses should be reduced by well lagging all pipes from the top of the economiser to the boiler. Quilted asbestos about 4 inches thick is usually found the best material for this purpose. Silicate of cotton in frames is also often used, but after a time, especially if anyone walks on it, the material comes out of the frames.*

Sometimes objection is taken to the use of an economiser where the feed water is highly charged with scale-forming matter, particularly if its temporary hardness is high. Such a water, however, is always objectionable for steam-raising purposes, and should be chemically treated before being used (see Chapter X.).

12. Berryman Feed Heater.—The latest form of the Berryman feed-water heater, as made by Messrs. Isaac Storey & Sons, is illustrated in Fig. 97. The lower chamber A is constructed of cast iron, being divided into two compartments by the vertical diaphragm B. The upper portion C is cylindrical, being built up of thin steel plates, as shown, and contains a series of **U**-shaped tubes D, which connect the two compartments of the lower chamber A. The exhaust steam enters one compartment, rises up one set of legs of the tubes D,

* For a discussion of various lagging materials see Art. 20, Chap. IX.

and descends into the other compartment, from which it issues. The feed water enters near the bottom of the cylindrical shell C and escapes through the pipe at the top.

13. Marshall Feed Heater.—The type of feed heater generally adopted by Messrs. Marshall, Sons & Co. is shown in Fig. 98. It consists of a wrought-steel barrel, provided with end covers which give access to the interior for cleaning purposes. The tube plates at the top and bottom are connected by a series of straight brass tubes, through which the delivery from the feed pump is passed on its way to the boiler. The exhaust steam is admitted near the bottom of the barrel, and passing around the tubes, leaves near the top. A drain and blow-off cock is arranged at the bottom of the heater in the position shown, and the steel barrel is encased with wood lagging.

14. Weir's Direct Contact Feed Heater.—A sectional view of this heater, which is used exclusively for marine work, is shown

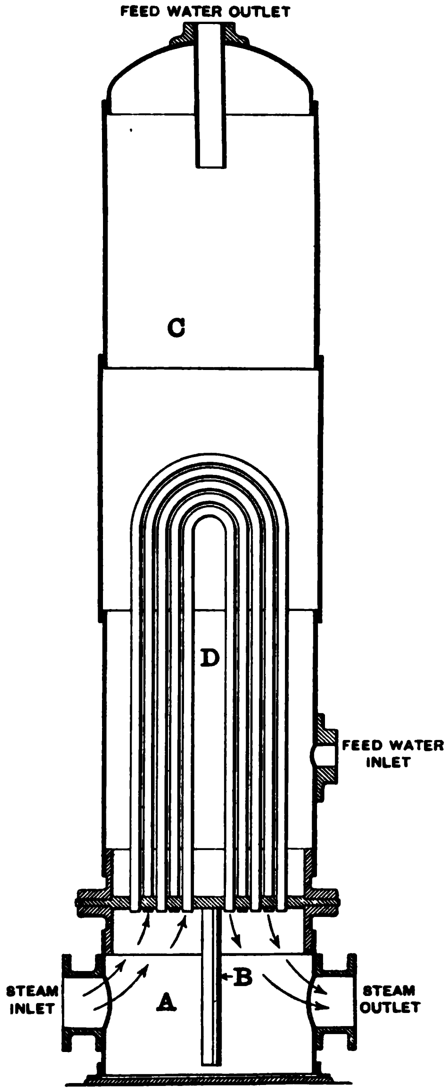


Fig. 97.—Improved Berryman feed-water heater.

in Fig. 99. The heating steam is taken from the low-pressure receiver of the main engine, and the exhaust of the auxiliary engines, such as feed pumps, electric light, fan engines, etc., is also led into the heater through the non-return valve B on the side of the apparatus. A perforated

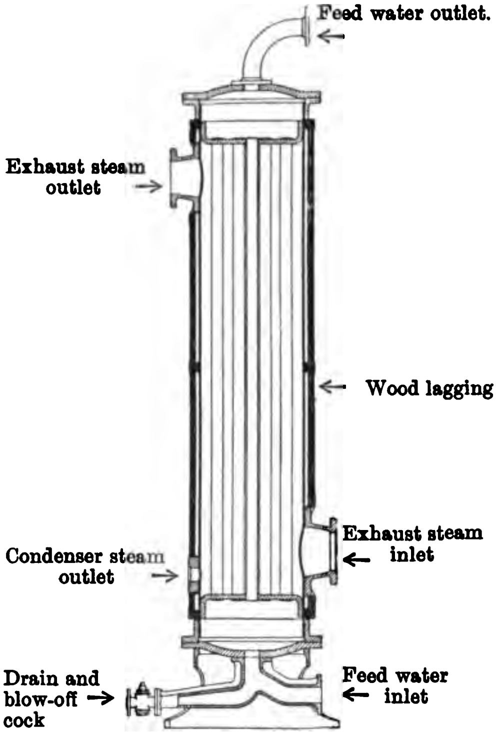


FIG. 98.—Marshall's feed-water heater.

cylindrical waist piece forming an annular steam space round the heater and surmounted by a conical spray piece, also perforated, is fitted to ensure the uniform mixing of the steam and water. The feed water is delivered from the feed pump into the heater through the spring-loaded valve D on the cover in a thin sheet, and is instantly heated by contact with the steam.

The water is further broken up into a fine spray by passing through the conical spray piece, thereby bringing the greatest possible surface into contact with the steam. As the pressure in the heater is generally much less than that of the entering

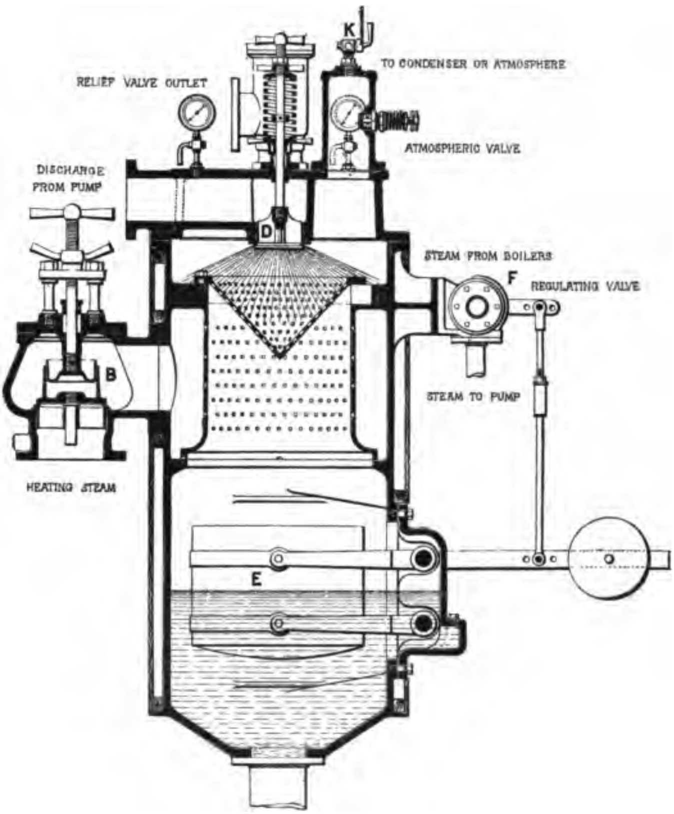


FIG. 99.—Weir's direct contact feed-water heater.

water, the effect of this lowering of the pressure, and sudden heating of the water, is to liberate the air in the water. This is removed to the condenser or to the atmosphere by a small cock K on the air vessel placed on the top of the heater. The feed

water is thus rendered practically non-corrosive, and falls to the bottom of the heater at the boiling temperature corresponding to the pressure.

The float E shown in the lower part of the heater is a pan, with water-tight bottom and sides, but open at the top. It is suspended on two levers, so as to move up and down with a parallel motion; the top lever spindle is carried through the door at one end, and is balanced by a lever and weight. The float is always full of water, and the weight is adjusted to balance when one-half is immersed in water. The weight lever is connected by a rod to another lever, which actuates the regulating valve F and controls the supply of steam to the pump drawing water from the heater. When the water in the heater rises, the float is raised and the regulating valve F opened, and when the water-level is lowered, the float falls with it and the valve is closed; the level of the water is thus automatically kept constant in the heater and the pumps are completely filled with water. A relief valve and two pressure gauges are fitted to the cover of the heater. One gauge shows the pressure in the heater, while the other shows the pressure of the ingoing water; the latter pressure should be about 15 to 25 pounds per square inch, and should be adjusted by the spring-loaded inlet valve D.

15. Feed Regulators.—With water-tube boilers having a small water capacity it is frequently desirable to install a feed water regulator, whose function is automatically to maintain the water-level in the boiler constant. Their use is confined to marine water-tube boilers chiefly of the small-tube type, the present tendency, however, being to use the type of boiler which does not depend on a feed regulator for its successful working.

16. Thornycroft Feed Regulator.—In connection with his water-tube boiler, Sir J. I. Thornycroft introduced his feed regulator shown in Fig. 100. It is an adaptation of the ordinary cistern ball-valve, in which the lever actuating the valve, and carrying the float and balance weight, is supported on a bell-crank lever. For purposes of adjustment the bell-crank is actuated by a connecting rod and a spindle passing

through a stuffing box. The height of the fulcrum of the lever can thus be altered, and the level of the water at which the valve shuts can be varied accordingly.

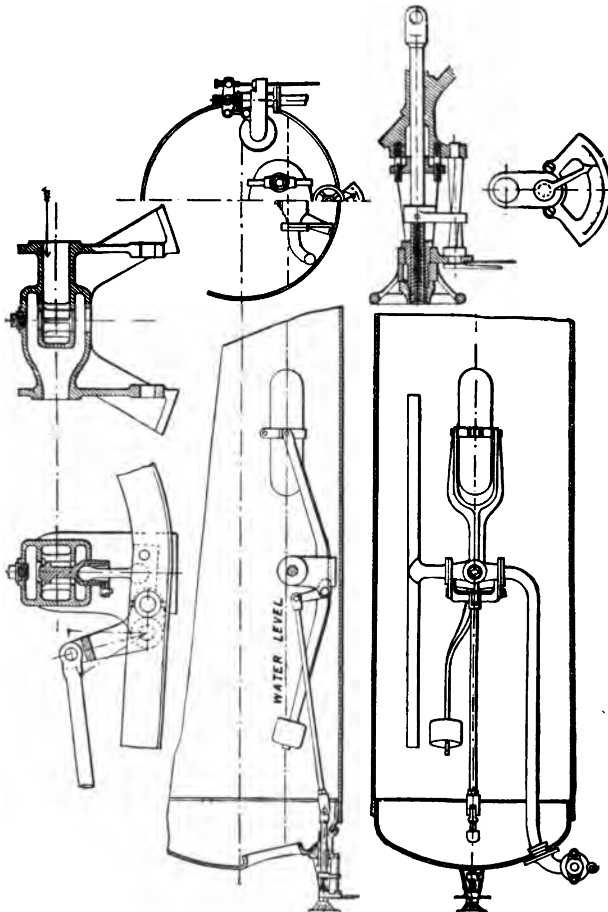


FIG. 100.—Thornycroft feed regulator.

The regulator is very useful for distributing the feed uniformly where two or more boilers are fed by the same feed pump. It is placed in the upper steam drum of the boiler, and

to the outlet is connected a perforated pipe, which distributes the feed water along the interior of the drum. To secure good results it is only necessary to maintain a steady pressure in the feed pipe of 40 or 50 pounds per square inch above the steam pressure, and to see that all the internal joints are tight.

17. Babcock and Wilcox Feed Regulator.—The arrangement of this regulator as applied to the shell of the steam drum of a boiler is illustrated in Fig. 101, whilst Figs. 102 and 103 show the arrangement applied to the end of the steam drum. The regulator consists of a hollow cylinder C, which may be rotated on its axis by means of a float and lever placed inside the boiler steam drum. The cylinder C has ports cut diagonally along its walls and works in a liner having corresponding ports. The liner is fitted in a casting or box which is attached to the boiler shell and has one branch for the feed check valve V, and another branch for a pipe leading to the non-return feed valve V.1. on the boiler shell. The cylinder C may be moved lengthways along the liner by means of a suitable adjusting screw and spindle, the spindle working in ball bearings so that friction is reduced to a minimum.

Attached to the spindle, between the adjusting wheel and gland, is a lever L and a rod, the latter extending to a position in the stokehold where it can be easily worked by hand from the firing level. The object of this rod and lever is to enable the attendant to ascertain that the float is moving freely when getting up steam, or to keep the float in the open or shut position should circumstances at any time so demand.

The action of the apparatus is as follows:—The feed water enters by the feed check valve V into the hollow cylinder C, and, if the water in the drum is low, passes through the ports of both cylinder and liner to the valve box, whence it travels through the pipe and non-return valve V.1. into the boiler.

The position of the cylindrical valve C is entirely regulated by the float in the boiler drum. When the level of the water in the boiler is low, the ports will be open, and as the water rises in the boiler and carries the float with it, the cylindrical valve will be moved round and the ports gradually closed. If the apparatus

be kept clean and in good condition, the friction of all the working parts being thereby a minimum, it will be sensitive enough to keep the opening of the ports just such as will maintain the

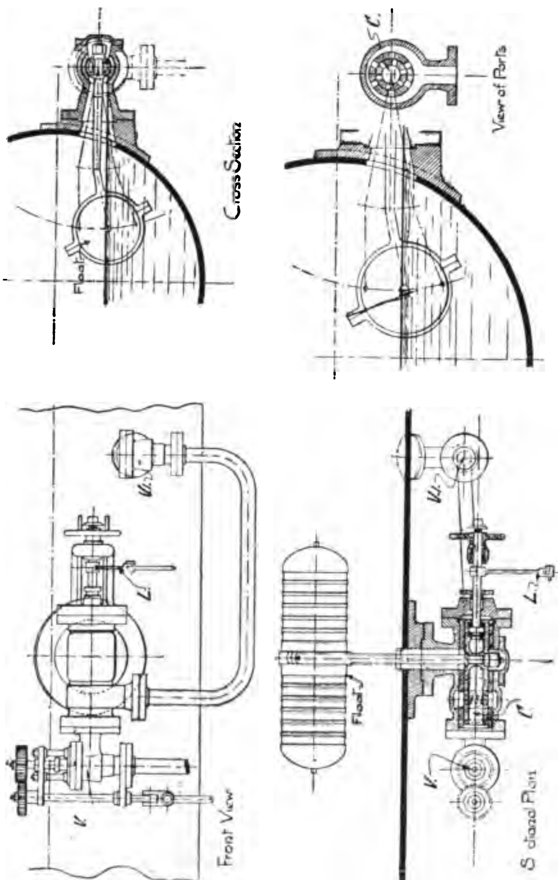
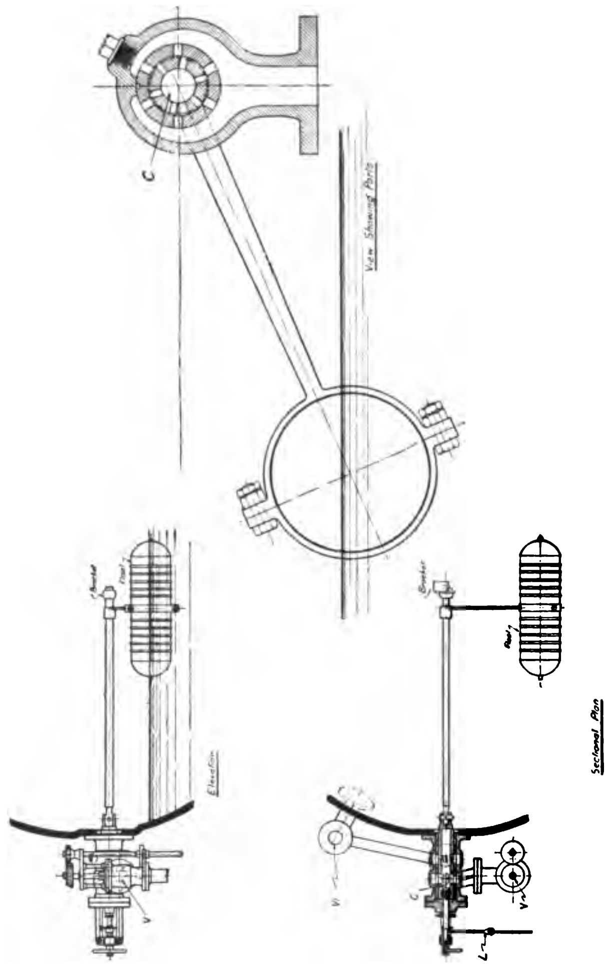


FIG. 101.—Babcock & Wilcox feed-water regulator applied to shell of drum.

water at a uniform level; there should never be a variation of more than 1 to 1½ inches in the water-level during the time the regulator is in action, and there should be no appreciable variation in the sensitiveness of the regulator when the boilers are worked

at a high or a low rate of evaporation, or even when the rate is changed suddenly.



FIGS. 102 and 103.—Babcock & Wilcox feed-water regulator applied to end of steam drum.

The height of the water-level in the boiler can be maintained at the position desired by moving the cylinder C lengthways along the liner. This is done by turning the handwheel on the

spindle attached to the cylinder, the travel of which, as the ports in both cylinder and liner are placed diagonally, will ensure an alteration in the point of cut-off relatively to the position of the float. It is usual to work the Babcock & Wilcox marine boiler with from 4 to 6 inches of water showing in the gauge glass, and the position of the regulator on the drum, and of the float, are so arranged that when the valve is in mid-position, this height of water will be regularly maintained, and any desired alteration of water-level in either direction can be effected by the adjustment of the handwheel on the spindle.

18. Reducing Valves.—A reducing valve is frequently fitted between the boiler and engine, and in some cases, as for instance with the Belleville boiler, it is absolutely necessary. It is essentially an accessory used in conjunction with water-tube boilers, because it enables one of the chief features of this type of boiler to be taken full advantage of, namely, the use of very high steam pressures, while at the same time the engines are protected from excessive pressure. In addition to this advantage it permits of the higher pressure in the boiler varying considerably without the pressure at the engines being affected, and also it slightly dries the steam, or if the steam is already dry, slightly superheats it. (Cp. the action of the throttling calorimeter of p. 376.)

In cases where the required pressure on the engine side of the valve is much below that on the boiler side, only a very small opening of the valve is necessary, and it becomes difficult to design a valve that will work satisfactorily. In conjunction with the large increase in the specific volume of the steam, its very high velocity as it passes through the valve is apt to cause undue wear of the contact faces.

19. Schäffer and Budenberg's Reducing Valve.—This valve is shown in section in Fig. 104. The valve is connected by means of the central spindle to a flexible diaphragm at the bottom. The central spindle, and therefore the valve, is pressed upwards by means of the horizontal lever, the requisite pressure being supplied by means of the external helical spring. It will be noticed that the high-pressure steam and the force exerted by

the spring through the lever act upwards on the valve, whilst the steam pressure on the reduced side of the valve acts downwards on it. The upward pressure exerted on the valve by the

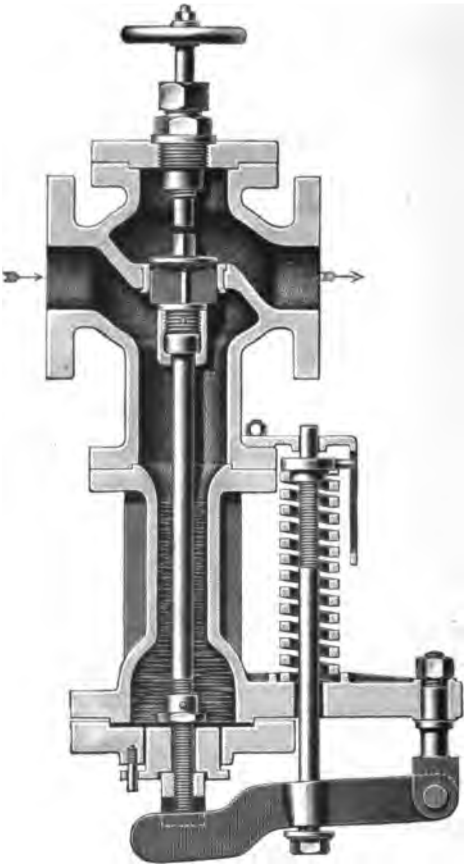


FIG. 104.—Schäffer and Budenberg's reducing valve (Auld type).

spring can be adjusted to neutralise the downward pressure on it and so render the reduced steam pressure independent of the original high pressure. The position of the stop above the valve

can be adjusted to give the valve area necessary for the particular drop in pressure required.

20. Steam Superheaters.—The increased economy arising from the use of superheated steam has resulted in many designs of superheater being placed on the market; in all cases, the steam is passed through a series of tubes which are surrounded by hot gases. In order to prevent any rise of pressure due to superheating, provision must be made for the expansion of the steam as its temperature rises. This expansion is provided for by the steam being drawn off and used, or, if no steam is used, by the safety-valve lifting. The study of the heat transmission through the heating surface of a superheater involves factors of peculiar interest, and although a considerable amount of research work has been carried out in order to determine the quantity of heat which may be transmitted through a metal plate having hot gases on one side and water on the other side (Chapter V.), very little appears to have been done when the water is replaced by steam. In any case the heat transmitted through a metal plate to steam will be less than to water; it therefore follows that a lower rate of heat transmission should be provided, and in order to prevent overheating of the superheater tubes the temperature of the hot gases should be as low as is practicable while still ensuring the amount of superheat required.

Superheaters may be divided broadly into two classes:—

1. Those in which the steam is superheated by the gases generated in the boiler furnace.
2. Those forming a separate unit having their own furnace, this type being known as the Independently-fired Superheater.

The general design of a superheater of the first class depends upon the type of boiler to which it is to be applied.

If a boiler is working under economical conditions, the difference between the temperature of the saturated steam and that of the flue gases leaving the boiler is not sufficient to superheat the steam to any appreciable extent unless the heating surface of the superheater is excessively large; hence a superheater should not be placed in the flue between the boiler and the chimney. On the other hand, if placed too close to the boiler

furnace, damage may result from the impact of the very hot gases. In the case of the Lancashire type of boiler the superheater is placed in the downtake at the back of the furnace tubes where the temperature of the gases is in the neighbourhood of 1000° F. The superheater must then be protected against overheating when steam is not passing through it in sufficient quantity to carry away the heat supplied by the furnace gases.

In addition to the above points the superheater should be constructed to admit of all its parts expanding and contracting freely without severe strains being put on any of the joints which might cause them to leak; flanged joints, when exposed to the furnace gases, should be avoided wherever possible. All superheaters arranged to give a high degree of superheat should have a bye-pass to the main steam pipe, so that steam can, when so desired, be taken direct from the boiler without superheating.

21. Babcock and Wilcox Integral Superheater.—This consists of solid drawn steel tubes G (Plate II.) about $1\frac{1}{2}$ inches in diameter bent into ⊂ shape and connected at each end by expanded joints to two wrought steel boxes or manifolds F and H. Saturated steam is taken from the boiler through the dry pipes N and the inlet tubes D into the superheater top box F; from here the steam passes through the superheater tubes G into the bottom box H. During its passage through these tubes G the steam is superheated, and is taken from the bottom box K through the outlet pipes J and the outlet cross pipe into the stop valve, from whence it passes to the engines. Hand holes are provided in the boxes F and H opposite the tubes for cleaning purposes. As will be seen from Plate II., the superheater is not subject to the direct impact of the fire as the furnace gases must first pass through the front portion of the boiler. There are no flanged joints, all the tubes G being expanded into the manifolds F and H, whilst freedom for expansion is provided by the tubes being free at one end, and by the manifolds not being rigidly connected with each other. Prevention against overheating during steam raising from cold water is ensured by the arrangement used for flooding with boiler water and using the superheater as part of the boiler heating surface.

The flooding arrangement consists of a pipe S in connection with the water space of the steam drum and two cocks B and C. By means of the larger cock B, water can enter through the pipe O into the lower box or manifold H and fill the superheater to the water level in the boiler. Any steam formed in the superheater tubes G is returned to the boiler steam drum through the two pipes D, which, when the superheater is at work, convey saturated steam into the upper manifold F. When steam is raised, and before opening the superheater stop valve and drawing off superheated steam from the boiler, the water is drained out of the superheater by closing the cock B and opening the smaller drain cock C. A sight glass A is attached to the drain outlet for observing when all the water has been blown out.

The makers do not recommend the regular use of the superheater as part of the boiler heating surface, but where it is necessary to provide for the temperature being regulated so as not to exceed a certain limit, they provide their patent superheat regulating valve. In this case the main stop valve is connected to the outlet flange of the regulating valve.

The regulating valve is composed of a steam chest, having an internal sector valve fixed on to a spindle, which passes through a gland and stuffing box in the cover. On this cover a quadrant is fixed, with the word "high" on one side and "low" on the other, meaning high and low temperatures of steam respectively, so that the lever on the end of the spindle may be firmly locked in any required position by the thumbscrew provided for the purpose.

The sector is of such a length on its circumferential surface that it obstructs the opening on either the saturated or superheated sides of the valve when one of the openings is entirely uncovered.

When the lever is at "high," the sector covers the saturated section, and allows only superheated steam to pass to the stop valve; and when at "low," it obstructs the superheater opening, letting the saturated steam through. At any intermediate position of the levers the sector partly covers each of the openings, and so allows both saturated and superheated steam

to pass into the steam chest, and to mix before passing to the stop valve.

The sector is made in such a way that, even if the saturated side is full open, a small quantity of steam passes through the superheater to keep up a circulation in it, but the superheat is neutralised by the much greater proportion of saturated steam.

22. Woodeson Integral Superheater.—This superheater consists of a series of steel tubes bent into a U-shape and expanded into headers. Its position in the boiler is between the front and middle sections of steam generating tubes as shown in Plate IV., the saturated steam being taken from the steam dome through the superheater tubes to the stop valve. To put the superheater out of action, a bye-pass valve and pipe is fitted between the two top headers. When this valve is opened, saturated steam passes directly from the steam dome through the bye-pass to the stop valve without entering the superheater. The superheater is of the non-flooding type, but a blow-off valve is fitted to the bottom header to drain off any condensed steam when the superheater is out of action.

23. Galloway Superheater.—The 1902 design of this superheater consists of a number of U tubes connected to and suspended from a wrought-steel plate which has two compartments milled out of it. The steam enters through the inlet at the top into one compartment, and after passing through the tubes, rises into the other compartment, from which it is taken off through the two outlets. The superheater is fitted in the downtake of the Lancashire boiler as illustrated in Fig. 105.

The 1910 design of superheater is shown fitted in the downtake in Fig. 106. All parts are of wrought steel and of circular form to ensure maximum strength. Handholes are fitted into the headers A and B opposite the tubes for cleaning and inspection purposes. The headers are raised above the brickwork with the special object of obtaining complete accessibility and first-class drainage, in addition to keeping them well away from the direct path of the gases. A cast-iron tray C with

double walls and fitted with a hood encloses the headers, the space between the tray walls being filled with a non-conducting composition to reduce radiation losses; the hood may be readily raised for examination. Saturated steam enters the header A and after passing through the tubes rises into the header B from which it is drawn off.

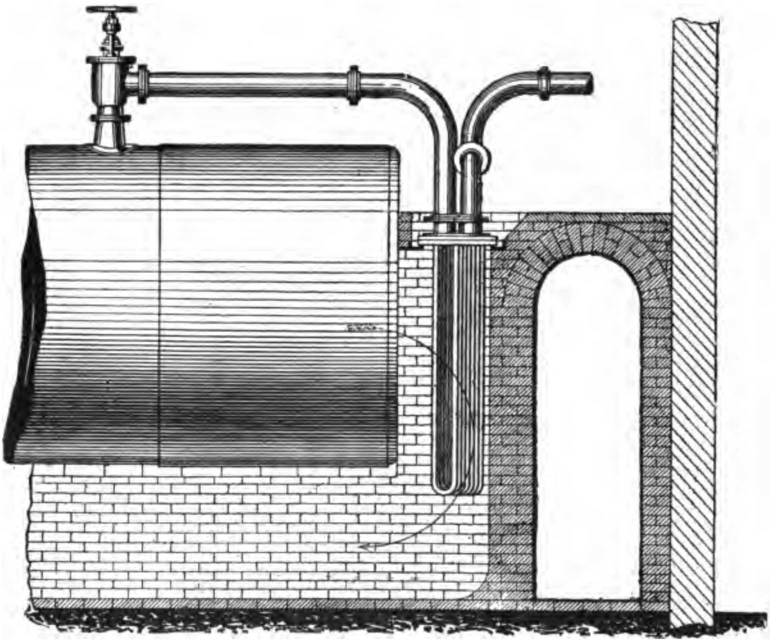


FIG. 105.—Galloway superheater fixed in downtake.

24. Sugden's Superheater.—This superheater consists of a series of steel U tubes expanded into wrought-steel boxes or headers at each end. Fig. 107 shows the superheater fitted in the downtake of a Lancashire boiler. The two headers run transversely across the top of the downtake, saturated steam being admitted into the header A furthest from the boiler. When saturated steam only is required the damper E is opened, the valves B and D are closed, and the steam taken directly

from the boiler through the valve C to the main steam pipe. When the superheater is in action, the valve C is closed, saturated steam passing through the valve B to the header A, from which it passes through the superheater tubes, the superheated steam passing through the valve D to the main steam pipe. The degree of superheat is regulated as follows: should the temperature of the steam exceed the desired value, it is

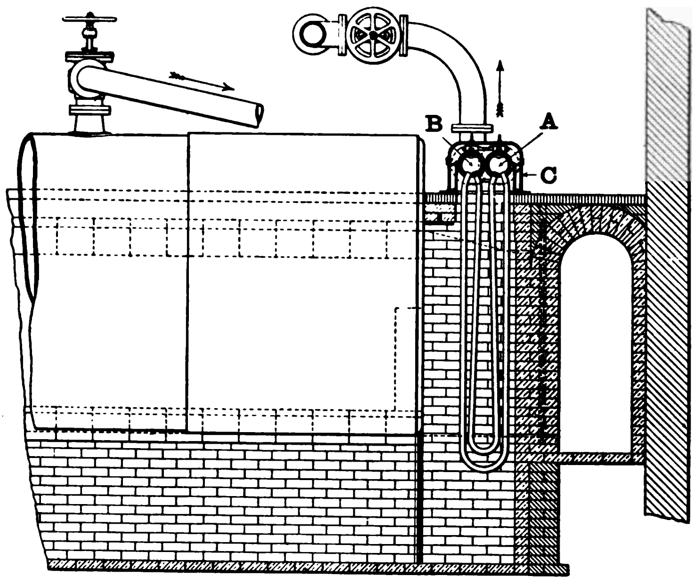


FIG. 106.—Galloway superheater fixed in downtake (1910 pattern).

reduced by partially opening the damper E, thus allowing a portion of the gases to go direct into the bottom flue and only a portion to impinge against the superheater tubes. When the damper is fully opened as shown on the right of Fig. 107 the superheater is completely isolated, but when the damper is closed the whole of the hot gases are compelled to pass through the superheater, the degree of superheat then being a maximum. This design requires a minimum width of downtake of 4 feet

6 inches and can usually only be adopted when the superheater is installed at the same time as the boiler.

A slightly modified arrangement is adopted in cases when the superheater is installed in an existing boiler, or when the minimum width above mentioned is not permissible. In this arrangement when the damper is closed the whole of the gases pass through the superheater as before; but with the damper fully opened, the superheater is not completely isolated, although the greater part of the gases go direct into the bottom flue.

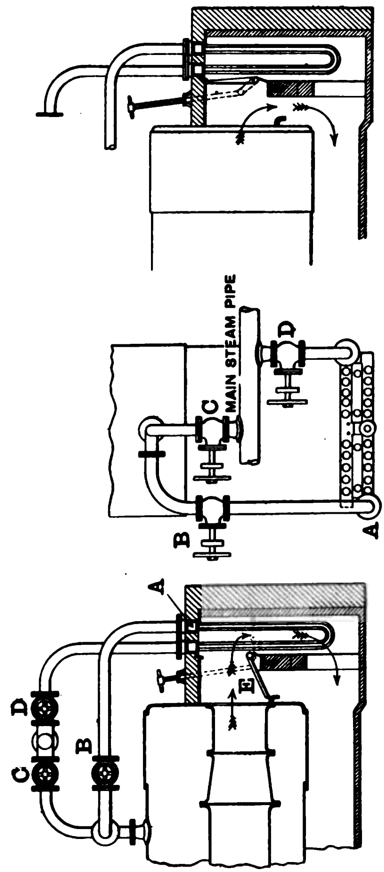


Fig. 107.—Arrangement of Sugden's superheater.

This type of superheater is also fitted to the Dryback boiler as shown in Fig. 108. The temperature of the gases in the downtake of this boiler is too high for the superheater to be fitted there unless the downtake is largely extended. Experiments show that the best position with this type of boiler is near the side walls as shown in Fig. 108. The superheater is made in two distinct sections, connected by means of crossover pipes, one section being placed near each side wall. The saturated steam from the boiler is directed by means of a three-way piece to the inlets of each section of the superheater,

whilst the outlets are connected, also by means of a three-way piece, to the main steam pipe.

25. Schmidt Superheater.—There are two distinct patterns of this superheater as applied to locomotives, viz.: the "Smoke-Box Type" and the "Smoke-Tube Type." Both types were invented by Dr. Schmidt at about the same time; until recently, the former has been favoured by the Prussian State

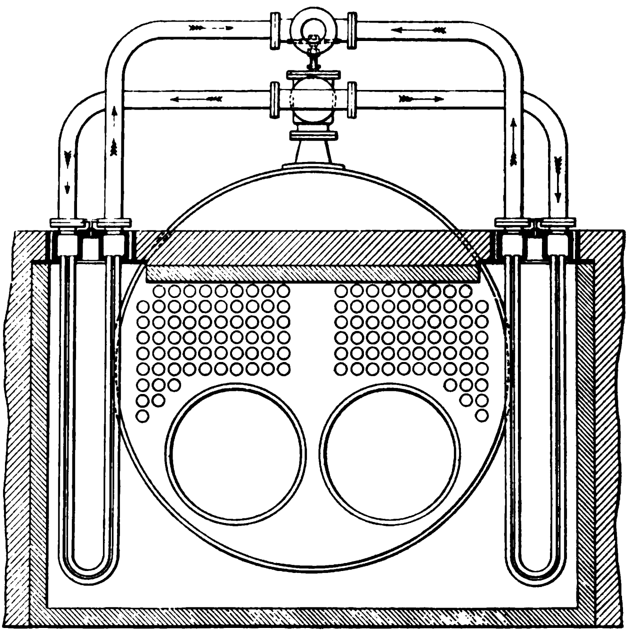


FIG. 108.—Sugden's superheater fitted to a Dryback boiler.

railways, whereas the latter type has, from the first, been preferred by most other railways. As regards fuel economy and efficiency there is no difference between the two types, but the smoke-tube superheater possesses important advantages over the smoke-box type, amongst which may be mentioned its more evenly distributed weight, greater simplicity, lightness, accessibility and lower cost. These advantages have proved to

be of such great value, that for all new work as well as for the Prussian State railways, the smoke-tube type only is being used. For this reason it is only proposed to describe here the smoke-tube superheater.

In this type (Fig. 109) the upper part of the boiler is fitted with from two to four rows of large seamless steel smoke tubes, which are expanded into the firebox and smoke-box tube plates in a special manner. These tubes are from 4 to $5\frac{1}{4}$ inches inside diameter (except at their firebox ends where the diameter

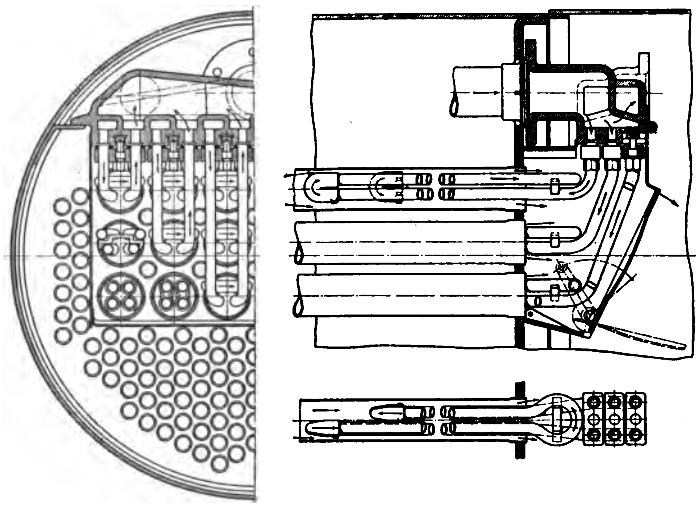


FIG. 109.—Schmidt's smoke-tube superheater (locomotive type).

is somewhat reduced), and inserted in each is a superheater element or section, consisting of two sets of pipes bent in the form of a U and connected at the smoke-box end to a header, thus forming a continuous double-looped tube through which the steam passes to and fro. The connections between the tubes on the firebox side are either made by U bends of cast steel or by welding. The open ends of each element extend into the smoke box, where they are bent upwards and expanded into a common flange which is secured to the face of the steam collector. The construction of the steam collector and its

connections to the steam pipe and valve chests are such that the steam has to pass through all the superheater tubes simultaneously on its way from the boiler to the engine cylinders.

The products of combustion from the firebox divide into two portions, one passing through the ordinary boiler tubes, and the other through the larger tubes. The heat contained in the gases is partly given up to the water surrounding those tubes, and partly to the steam contained in the enclosed superheating elements. The flow of heat through the large tubes is controlled by dampers of different designs, hinged or pivoted below the steam collector in the smoke box. As long as the regulator is shut, these dampers are kept closed by either a counterweight or spring; but immediately steam is turned on by opening the regulator, they are opened simultaneously by means of a piston working in a small automatic cylinder. Thus, whilst getting up steam, or whenever the regulator is closed, and the superheating tubes are not cooled by steam, none of the hot products of combustion from the firebox can pass through the large smoke tubes; the superheating tubes are by this means prevented from being overheated. If desired, the superheater dampers can be worked by hand from the footplate independently of the automatic cylinders, so that any required degree of superheat can be attained; when the dampers are open, they permit of a clear view through the superheater.

From tests made on this type of superheater it appears that the amount of superheat does not increase directly in proportion to the draught or rate of combustion, but falls off at the higher rates of working, owing to the gases leaving the superheater tubes being at a lower temperature than those leaving the ordinary boiler tubes. This is probably due to a reduction in the proportion of the gases passing through the superheater tubes consequent upon a greater resistance to flow at high rates of working.*

26. Schmidt's Marine Superheater.—The superheater consists of collector pipes usually arranged vertically between the nests of tubes, as shown in Fig. 110. The collector pipes

* See *Engineering*, July 27, 1912.

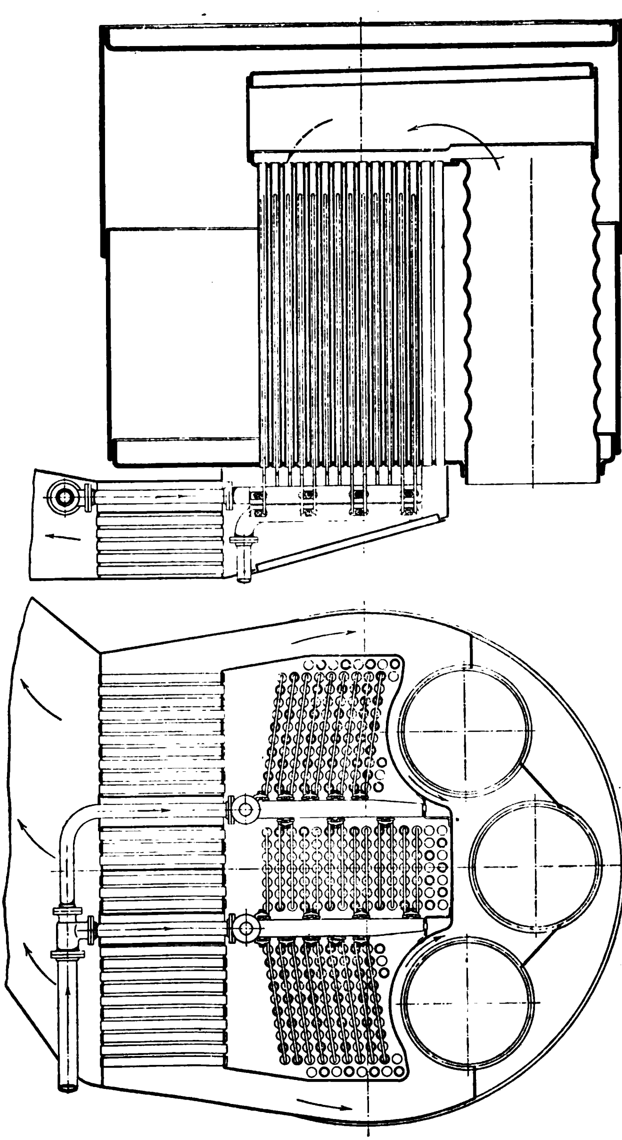


FIG. 110.—Schmidt's marine smoke-tube superheater.

are in two parts, one taking the saturated steam from the boiler and the other delivering the superheated steam to the engines. From these collectors the superheater tubes are led through the smoke tubes as in the locomotive type. The ends of the superheater tubes where they join the collectors have a collar which forms a joint, and are held in place by a cast-steel dog secured with a single bolt, as shown in Fig. 111. If by any chance one of the superheater tubes should leak, the leakage can be easily detected. By unscrewing two nuts the defective part can be removed, a stopper placed in the hole, and the boilers may continue steaming, the whole operation only occupying a few minutes. When removing one of the superheater elements it is only necessary to close the stop valve and blow out the steam contained in the tubes and collectors, the latter being fitted with a valve for this purpose and for drainage. The superheater-tubes are cleaned with either a steam or compressed air jet in both types of superheaters.*

27. Independently Fired Superheaters.—In large installations, with many boilers working together, an independently fired superheater is frequently preferred to an integral superheater with each boiler. The degree of superheat is more easily regulated in an independently fired superheater than in an integral superheater, but the separate furnace of the former means more radiation loss and more labour, so that it is only warranted when there is a large number of boilers. It is not an easy matter either to design or to construct a separately fired superheater to give satisfaction and to be both durable and free from leakage, on account of the high temperature of the furnace gases (see Art. 20). The superheater should be made entirely of wrought steel, with thick seamless tubes of small diameter, through which the steam should circulate rapidly. The tubes should not be subjected to the direct impact of the furnace gases, but should be in a chamber filled evenly with completely burnt gases whose temperature is as low as practicable to ensure the final superheat required. The areas through which

* See paper by A. F. White on "Marine Engines and Superheated Steam," read before the Institute of Marine Engineers on Nov. 15, 1909.

the steam passes should be considerably larger than the steam inlet and outlet connections to avoid drop in pressure, whilst the entire construction should offer the greatest freedom for expansion and contraction.

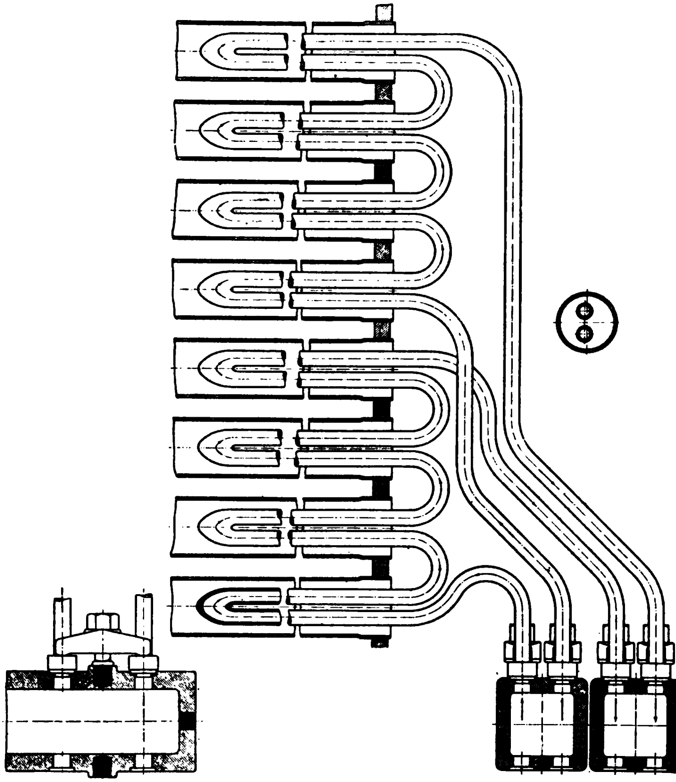


FIG. 111.—Details of Schmidt marine smoke-tube superheater.

28. Babcock and Wilcox Independently Fired Superheater.—In this superheater the steel ∩ tubes are expanded at their lower ends into a series of transverse manifolds or junction boxes (Fig. 112). The manifolds A leading from the saturated steam inlet on the left of the superheater, are divided into two compartments by means of

diaphragms. These diaphragms are shown in the sectional plan

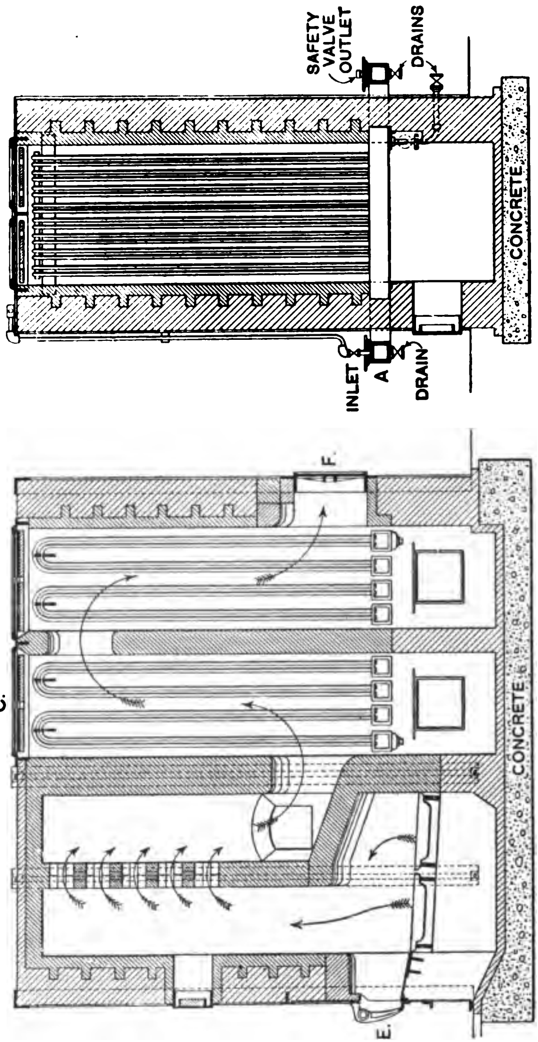
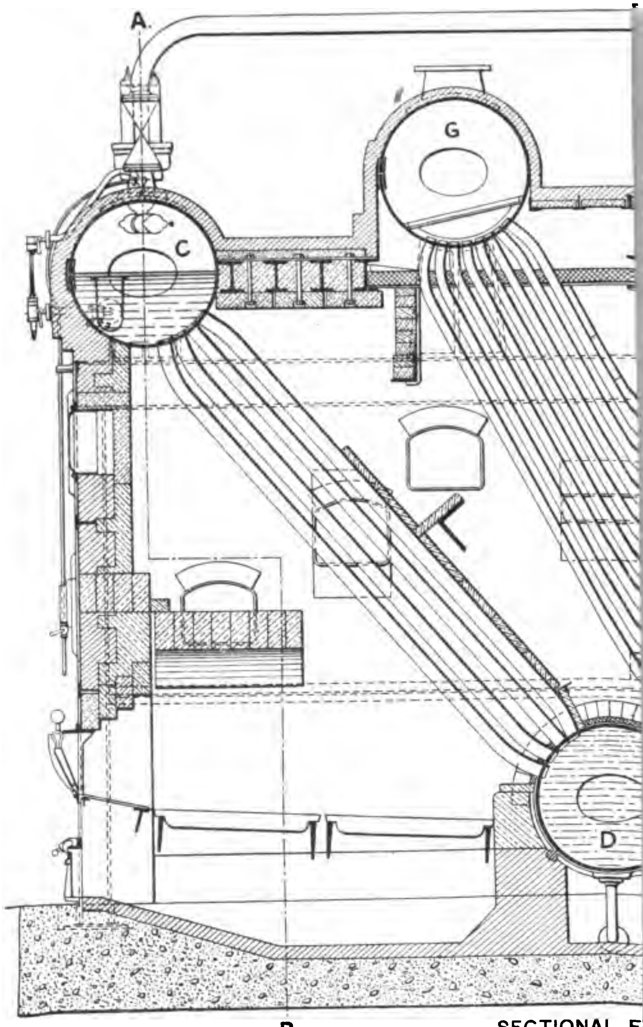


Fig. 112.—Babcock & Wilcox independently fired superheater.

view (Fig. 112A) at B, and they compel the steam to follow the path shown by the arrows in that view. In order to protect



SECTIONAL E

PLATE VI

the tubes from the fierce heat of the fire, the furnace is arranged as a "gas furnace;" the tubes are surrounded by the burnt gases, and are not in direct contact with the flames, thereby fulfilling the important condition mentioned in Arts. 20 and 27. The hot gases, of course, leave the superheater at a higher temperature than that of the superheated steam, but their waste heat—which would otherwise be lost—may be utilised by leading them into an adjacent boiler or economiser.

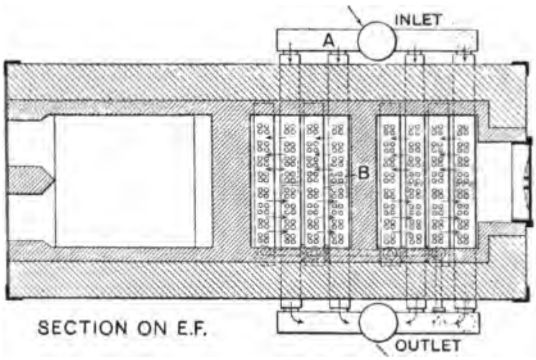


FIG. 112A.—Babcock & Wilcox independently fired superheater (plan).

29. Stirling Independently Fired Superheater.—The general arrangement of this superheater is somewhat similar to the five-drum type of Stirling water-tube boiler, its construction being clearly shown in Plate VI. The front top steam and water drum C, and the front bottom water drum D with their connecting water tubes, constitute one section of a Stirling boiler through which the furnace gases pass on their way to the superheater portion. By this means, overheating of the superheater tubes is guarded against, the temperature of the gases being reduced to about 1300° F. before entering the superheater portion and some of their heat being utilised in generating saturated steam in the front boiler section.

The main supply of saturated steam is admitted into the top rear steam drum E through the stop valve shown. After leaving the steam drum E, the steam passes up and down various sections of the two banks of superheater tubes until it

is finally discharged into the top front steam drum G. The three steam drums E, F and G are fitted with baffle boxes, as shown, there being two baffle boxes, K and M, in the top rear drum E, two, N and P, in the top front steam drum G, and four, H, L, O and Q, in the bottom steam drum F; the function of these baffle boxes is to compel the steam to take a sinuous path through the superheater.

The key plan (Fig. 113) shows the circuit made by the steam. From the top rear inlet steam drum E the steam passes down some of the rear bank of superheater tubes into the baffle box H. From this box it ascends into the baffle box

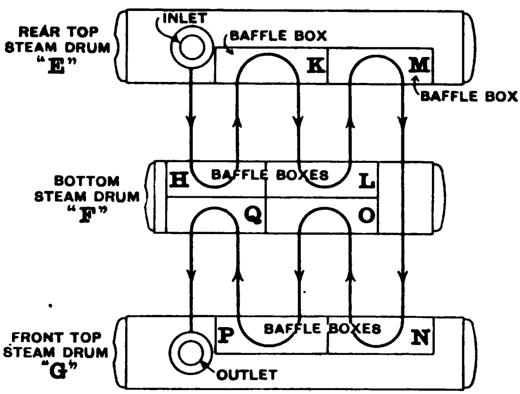


FIG. 113.—Key plan of Stirling independently fired superheater (Plate VI.) showing circuit of steam.

K in the steam drum E, and then descends into the box L, from which it again ascends into the second box M in the steam drum E. From M, the steam passes to the baffle box N in the front top steam drum G, and then descends some of the middle bank of tubes into the baffle box O in the bottom steam drum F, from which it ascends into the box P and then descends again into the box Q, from which it finally rises into the front top steam drum G. The superheated steam is withdrawn from the drum G through the main superheater stop valve.

Feed water is admitted into the top steam and water drum

C of the boiler portion by means of a feed trough, as in the Stirling boiler already described (Art. 7, Chap. VII.), and the saturated steam generated in the boiler portion C D is taken through the usual antipriming pipe, along the equalising pipe arranged above the superheater, into the rear top steam drum E, where it mixes with the main inlet saturated steam. A safety valve is fitted on both the steam drums C and E. Blow-off cocks are fitted to the water drum D and the bottom steam drum F, and drain cocks are fitted on each of the four baffle boxes in F.

The firegrate is arranged across the superheater immediately in front of the boiler portion, the furnace gases travelling upwards among the steam generating tubes, over the firebrick baffle arranged between the boiler portion and the superheater proper, downwards among the first bank of superheater tubes, and upwards among the rear bank of superheater tubes and then off to the chimney.

In this type of superheater, about 50 per cent. of the heat absorbed is used to generate steam in the boiler portion, thereby reducing (without loss of heat) the temperature of the gases to a safe value (about 1300° F.) before entering the superheater portion. The remainder of the heat is absorbed by the superheater proper, which superheats both the steam from the main boiler plant and that produced in its own boiler portion.

Independently fired superheaters of this type can be furnished to superheat up to 200,000 pounds of steam per hour with any degree of superheat required.

CHAPTER IX

BOILER MOUNTINGS AND STEAM PIPES

1. Stop Valves.—The first essential for a stop valve is that when closed it should be steam and water tight. Leakage must therefore be avoided, and even with careful design this is not always an easy matter. Leakage is commonly due to the unequal expansion of the various parts of the valve, and although when cold the valve may bear equally all round its seat, it will not do so when hot. When employed with saturated steam of comparatively low pressure and temperature, the body of the valve is usually made of cast iron with a cast-iron valve having a gun-metal seat. At a temperature of about 400° F., however, gun metal is considerably softer than when cold, and there is the danger of the valve seat becoming grooved, with the result that it is not possible to maintain tightness when closed.

Ordinary cast-iron valves with gun-metal seats are unsuitable for use with superheated steam, since the high working temperature exaggerates the above difficulties. The coefficients of expansion of cast-iron and gun-metal differ appreciably, and at high temperatures (in some cases the temperature at a superheater stop valve may be as high as 1000° F.), the unequal expansion of the valve body and seat, together with the great and unequal reduction in the strength of the materials (see Art. 13, Chap. I.), results in the valve seat becoming quite loose, although in the first instance the seat may have been tight enough to give complete satisfaction when used with low-pressure (and therefore low-temperature) saturated steam.

The valve chests and covers of superheater stop valves should be made of cast steel of reliable quality, and the valves, seats and spindles of nickel steel. The contact faces of the

valve and seat should be flat and as narrow as is practicable; for a large valve of, say, 10 inches diameter, the width of the bearing surface should not exceed about $\frac{3}{16}$ths of an inch.

When under steam, the body of a cast-iron or steel valve must necessarily expand, but also in most cases there is a certain amount of distortion as well as of expansion, and this increases with the rise of temperature. If therefore the main casting which supports the valve and its seat fails to retain its shape under superheated steam, even though the distortion may be very slight, leakage will ensue which will soon groove the seat and valve faces. To prevent this, it must be possible to apply sufficient force so as to ensure the surface of the valve and seat remaining in close contact under all conditions.

The requirements for an ideal stop-valve are therefore as follows:—

(1) The distortion should be reduced to a minimum. This is secured by so designing the valve-chest, etc., that the castings are as simple and as symmetrical as possible. There should be no abrupt or great differences in the thickness of the metal walls which might set up unequal expansion under high temperatures.

(2) Whatever amount of distortion still remains, the force necessary to counteract it should not call for any great physical effort. In some cases large valves are arranged to be operated by gearing, and very large valves, especially in electric light stations, are sometimes worked through worm-gearing by electric motors controlled from a distance. But apart from such cases, when operated by hand, it is necessary to make the hand-wheel of the valve of such large diameter that, with the leverage so obtained, a comparatively small effort applied to the circumference of the wheel will suffice to force the valve and seat together when they are hot and slightly distorted. To close the valve and render it tight against steam at a pressure of 200 lbs. per square inch is a very different matter from closing it when cold against the same water pressure.

The large diameter of hand-wheel above recommended will, of course, entail a proportionately heavy spindle, etc.

2. Hopkinson-Ferranti Stop Valve.—This valve, made

by Messrs. J. Hopkinson & Co., Ltd., is shown in Fig. 114, and is designed with working parts half the diameter of the ordinary type of stop valve to pass the same amount of steam. As shown in Fig. 114, the valve is closed. The valve seats A and the valve discs B are made of Hopkinson's "Platnam" metal to withstand the action of high-pressure and temperature superheated steam. The valve discs B are pressed into con-

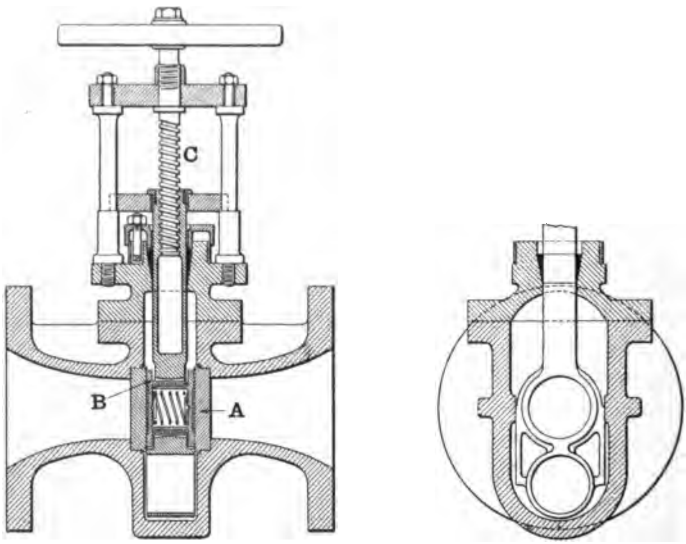


FIG. 114.—Hopkinson-Ferranti patent stop valve.

tact with the seats A by means of the compressed helical spring shown, and are free to rotate on their axes, so that in opening and closing they slide upon their seats with a flexible pressure which cleans the faces and prevents them from being cut and scored by grit. When the valve is open and working, the seats are sheltered from the scoring action of the steam flowing through the valve; the valve is opened by raising the valve discs by means of the screw C, actuated from the hand-wheel at the top.

The valve works on the well-known principle as explained

by Bernoulli's theorem.* Boiler steam enters at one end of the converging nozzle at a certain pressure, temperature, and velocity, and flows along the specially designed nozzle towards the valve. Since by Bernoulli's theorem the sum of the pressure energy and kinetic energy of the steam remains constant, and each cross section of the nozzle passes the same weight of steam per second, it follows that the pressure falls, and the velocity of the steam increases, as it flows towards the valve, the velocity being a maximum and the pressure a minimum at the valve. In other words, some of the pressure energy of the steam at inlet is converted into kinetic energy at the valve. Having passed through the valve, the steam enters the diverging nozzle in which the converse operations take place, the pressure increasing whilst the velocity decreases, until at the outlet end of this nozzle the condition of the steam as regards pressure, temperature and velocity is nearly the same as at inlet. The best shape of the nozzles has been determined experimentally, the greatest cross section being 4 times that at the valve; in the valve shown in Fig. 114, the diameter of the valve is 3 inches, and that at inlet and outlet is 6 inches. If there were no losses due to friction and eddies, the steam would leave the valve with exactly the same pressure and velocity as at entry, and in order to reduce these losses to a minimum the nozzles are made very smooth with the proper curvature. This valve will pass the same amount of steam as an ordinary valve 6 inches diameter, and is obviously very much smaller.

3. Combined Stop and Isolating Valve.—When several boilers are delivering steam into a common steam range, in addition to the stop valve an automatic isolating valve is sometimes used. In the event of a steam pipe bursting, this completely isolates the boiler, and by so doing prevents a dangerous rush of steam through the burst pipe. A combined stop and isolating valve made by Messrs. Schäffer & Budenberg is illustrated in Fig. 115. In this valve the automatic closing mechanism is entirely independent of the stop valve.

* See F. C. Lea's *Hydraulics*, 2nd edit., p. 39. London: Edward Arnold.

The latter is only opened when the engine is started, and closed when steam is no longer required and the engine stopped. The stop valve A is opened or closed by rotating the screwed spindle B by means of the hand-wheel keyed to it. The automatic isolating valve C is placed in a chamber immediately below the stop valve A, and is carried on the bell-crank lever D, which is so adjusted, by the spring E outside the

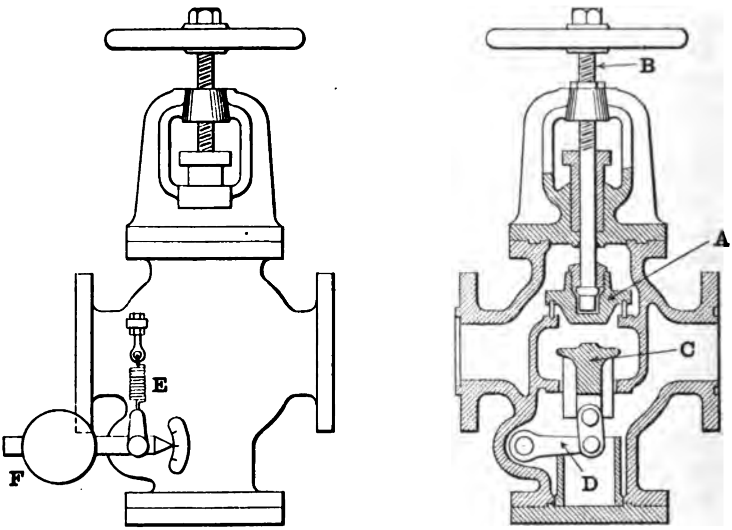


FIG. 115.—Combined stop and isolating valve.

casing, that the valve C, under working conditions, is maintained in its middle position as shown.

The principal feature of the valve consists in its closing automatically in either direction in case of any accident to the steam pipe or to one of a battery of boilers feeding the same steam main. If the steam pipe to the left of the valve in Fig. 115 bursts, the rush of steam from the right lifts the valve C upwards and closes the steam passage, whereas if the pipe to the right of the valve bursts, the rush of steam from the left forces the valve C downwards and so closes the steam

passage. The automatic valve C at once resumes its normal or middle position as soon as the upper main stop valve A is shut. A scale is provided on the outside of the valve showing whether the automatic valve C is open or shut in the upward or downward position. The efficiency of the valves can be tested at any time by means of the outer lever carrying the balance weight F.

4. Safety Valves.—All boilers should, whenever possible, be fitted with two safety valves, each having its own direct connection to the boiler. In Lancashire and Cornish boilers it is the usual practice for one valve to open when the steam pressure exceeds the working pressure for which the boiler is designed, whilst the other valve opens *in addition* when the water level falls too low, this valve being called the high-steam and low-water valve. There are three types of safety valve in common use, viz., the lever safety valve, the dead-weight safety valve, and the spring-loaded safety valve. The Board of Trade has laid down certain rules to govern the design and working of safety valves on marine boilers, viz.:—

1. One of the two valves with which boilers are fitted must be of the lock-up type, that is to say, it must be out of the control of the attendant when steam is up.

2. The area of each valve shall be proportioned to the area of the firegrate according to the rule

$$A = \frac{37 \cdot 5 \times G}{P}$$

where A = area of valve in square inches.
G = area of firegrate in square feet.
P = working steam pressure in pounds per square inch *absolute*.

In any case the diameter of the valve must not be less than 2 inches, and the maximum lift of the valve should be equal to one-fourth of the diameter.

3. After the boiler has been running under full load, with the stop valve closed and the feed water shut off for at least 20 minutes, the steam pressure in the boiler shall not rise more than 10 per cent. above the loaded pressure; *i.e.* if the boiler

working pressure is 100 pounds per square inch, the pressure must not rise above 110 pounds per square inch.

4. The safety valves on each boiler must be provided with lifting gear so arranged that they can be eased together and operated by hand, either from the stokehold or engine room.

5. **Lever Safety Valve.**—This type of valve consists of a lever (Fig. 116) pivoted at the end F. The valve is attached to the lever at some point V close to F. The centre of gravity of the lever is at point G, and the valve is held on its seat against the upward steam pressure by the combined action of the weight W hung from some point A on the lever, the weight of the lever acting at G, and the weight of the valve and its spindle acting at V. The weight W and the distance AF are

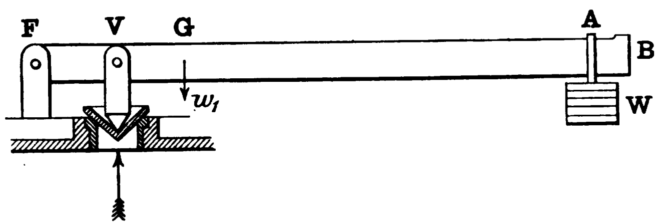


FIG. 116.—Diagram of lever safety valve.

so adjusted that when the steam pressure acting *upwards* on the valve reaches a certain value, it overcomes the *downward* force exerted on the valve by the weight W. The result is that the valve opens and steam escapes until its pressure falls to the working value when the valve closes again.

Let d = diameter of the valve in inches,

p = working pressure in pounds per square inch gauge,

w_1 = weight of the lever acting downward at G,

w_2 = weight of the valve acting downward at V.

Then, taking moments about the fulcrum F, we have

$$\left(p \times \frac{\pi d^2}{4}\right) \times \mathrm{FV} = (w_2 \times \mathrm{FV}) + (w_1 \times \mathrm{FG}) + (\mathrm{W} \times \mathrm{FA})$$

Example.—A lever safety valve is to blow off when the steam pressure is 180 pounds per square inch above atmospheric

(*i.e.* gauge). The valve is 3 inches diameter and weighs 2 pounds, being attached to the lever 4 inches from the fulcrum. The weight of the lever is 6 pounds, and its centre of gravity is 10 inches from the fulcrum. What weight must be hung on the lever 36 inches from the fulcrum?

Taking moments about the fulcrum

$$\left(180 \times \frac{\pi}{4} \times 3^2\right) \times 4 = (2 \times 4) + (6 \times 10) + (W \times 36)$$
$$\pi \times 9 \times 180 = 8 + 60 + 36W$$
$$5089 = 68 + 36W$$
$$36W = 5089 - 68 = 5021$$
$$W = \frac{5021}{36} = 139 \text{ pounds.}$$

A lever safety valve of the enclosed type is shown in Fig. 117. The flange A is bolted to the boiler seating block, while to the

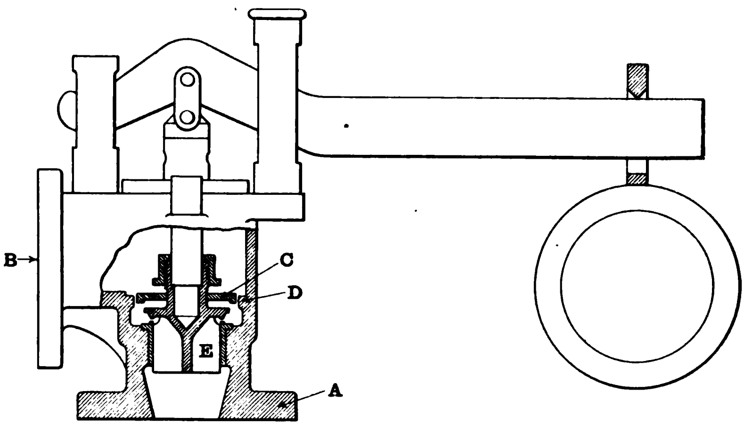


FIG. 117.—Lever safety valve, enclosed type.

flange B is coupled a pipe through which the steam is led away when it blows off. In this way the inconvenience caused by the steam escaping directly into the boiler house is obviated. An adjustable disc C of larger diameter than that of the valve is arranged above the valve proper E, whilst a lip D is formed opposite, on the inside of the casing, leaving a narrow annular slit between the disc and lip.

Under ordinary circumstances, the steam which blows off is allowed to escape through the annular space between the disc and lip, and the valve E will close at the same pressure at which it commenced to blow off. If, however, the pressure

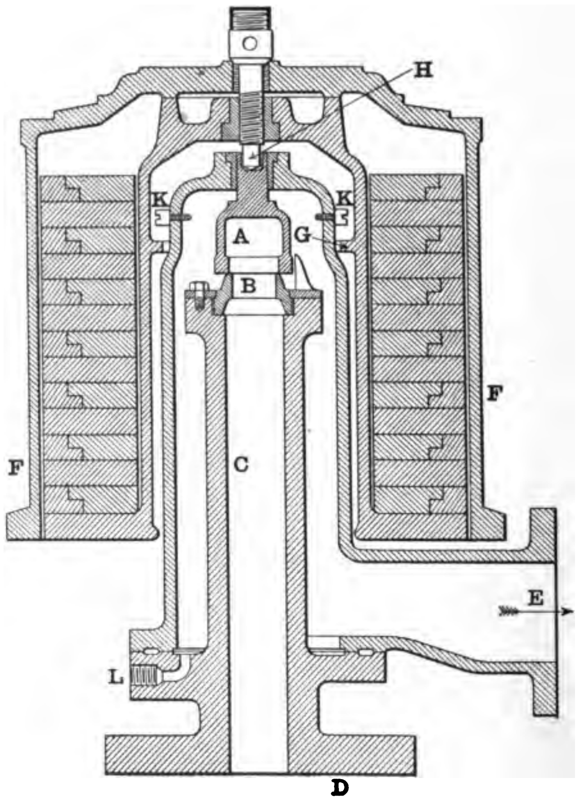


FIG. 118.—Hopkinson's patent "Ipsed" dead-weight safety valve.

rises still further, the lift of the valve E increases, and an accumulation of pressure results in the annular space under the disc, causing the valve to open smoothly to its full lift. The valve closes smoothly again as soon as the surplus steam has

been discharged. The valve is generally adjusted to ensure the high lift commencing as soon as the pressure is increased by about 3 pounds per square inch; at an increase of about 8 pounds the valve is practically fully open. In order to prevent overloading of the valve whereby it would remain closed after the steam pressure had reached a value higher than that at which it should open, the dead weight should be fixed as close as possible to the *end* of the lever.

6. Dead Weight Safety Valve.—Fig. 118 shows a safety valve of this type made by Messrs. Hopkinson & Co. The valve A rests on a seat B, which is mounted on the top of the pipe C, whose flange D is bolted to the boiler seating block. The valve shown is of the enclosed type, so that when the valve lifts the steam is discharged through a pipe coupled up to the flange E. The valve is held on its seat by means of the required number of disc weights contained in the outer annular casing F. It will be noticed that the valve is pendulously weighted, the dead weight being applied to the top of the valve by means of the pin H, which has a conical end bearing upon the valve, and further, the greater part of the dead weight lies below the valve seat so that there is no possibility of the weight being displaced in a lateral direction. An annular ring G is cast on the inside of the cover F, the lift of the valve being controlled by this ring coming against the set screws K. A drain cock is fixed to the connection at L in order to drain away any steam that may condense in the discharge pipe.

7. Dead-Loaded Spring Safety Valve.—In this type of valve the downward force holding the valve on its seat is supplied by means of a compressed helical spring. In all cases where the boiler is subject to vibrations, as in locomotive and marine boilers, the dead weight and the lever safety valves are inadmissible, and the spring-loaded valve must be used. Fig. 119 shows a spring-loaded marine type valve of the Board of Trade pattern. It will be noticed that the tops of the valves are provided with an extended lip of larger diameter than the seating, as in Fig. 117, in order to secure a rapid lift of the valve as soon as it commences to blow.

In all valves of the enclosed type a drain cock should be fitted in the lowest part of the valve casing, and the waste pipe carrying the blow-off steam should be arranged in such a manner that the discharge is visible.

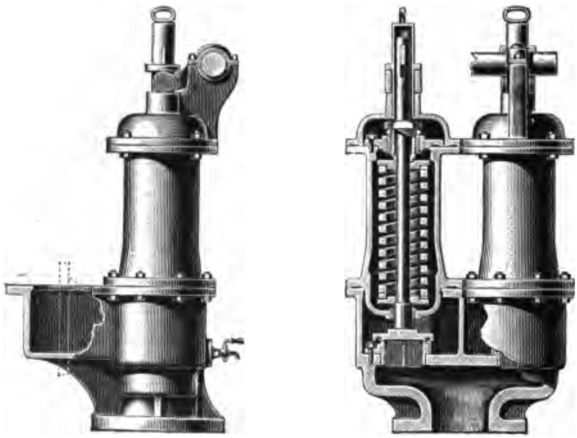


FIG. 119.—Marine safety valve by Messrs. Schäffer & Budenberg.

8. Combined High Steam and Low Water Safety Valve.—An example of this type of valve which is in extensive use is the Hopkinson "Duad" Safety Valve for High Steam and Low Water. This valve, originally brought out by Messrs. J. Hopkinson & Co., Ltd., in 1852, comprises a main valve A (Fig. 120) loaded by means of a lever and weight outside the boiler and resting on a "Platnam" seat let into the valve block, which is bolted to the riveting base B on the boiler shell. The interior of this main valve is formed as a seating to receive a second valve of hemispherical shape loaded by means of a dead-weight C inside the boiler attached to its spindle G. The combined valves are designed to blow off together for high steam.

A lever inside the boiler, with a fulcrum D attached to the riveting base, is loaded at one end by a float E of vitrified earthenware, the weight of this being partly balanced by a cast-iron weight F at the other end of the lever. The float rises

and falls with the surface of the water in the boiler. If the water sinks below a predetermined level, the sinking of the float causes two points on the lever to come into contact with a screwed washer on the spindle G of the small inner valve,

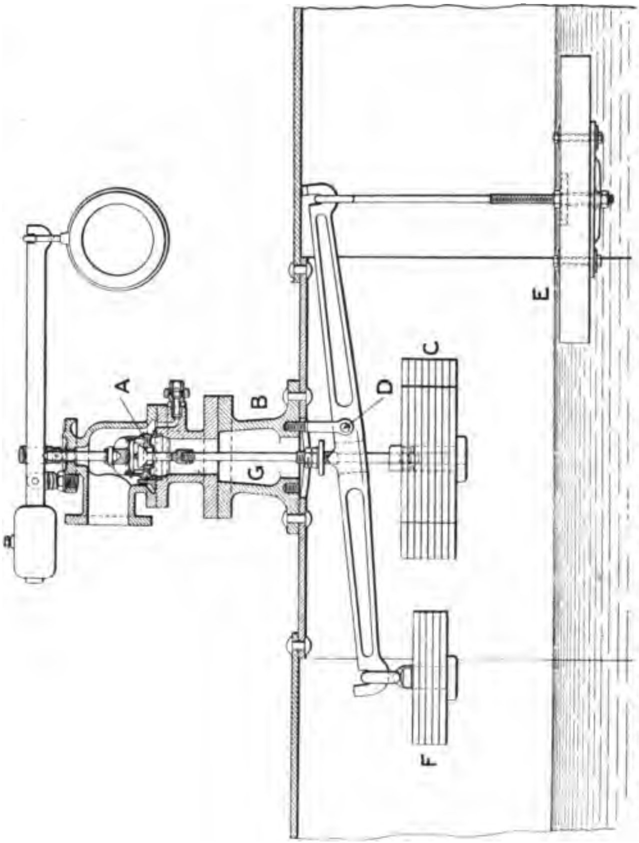


FIG. 120.—Hopkinson's "Duad" safety valve for high steam and low water.

lifting this from its seat. The blowing-off of this small valve gives the alarm for low water. In some designs, the steam blowing off when the water level is too low is made to work a steam whistle, thus giving audible proof of the low water level and constituting a "low-water alarm."

The dead weight on the lever loading the main valve is arranged to be of precisely the same number of pounds as the desired blow-off pressure, and maintains one fixed position on the lever. The entire apparatus is designed to permit of easy examination or inspection of every part when the boiler is laid off for cleaning.

Prior to the introduction of this valve, boiler explosions due to low water were unfortunately only too frequent, but nowadays this or a similar device is universally fitted on all Lancashire boilers, thereby preventing the serious loss of life and property from this source.

9. Fusible Plugs.—A low-water safety valve as contrasted with a low-water alarm actually lowers the steam pressure, whereas escape of steam through low-water alarms is practically negligible, since it is only sufficient to work the whistle. As already mentioned (Art. 8) the high steam and low-water safety valve is usually fitted to the Lancashire type of boiler, but in addition thereto there is often a *fusible plug*, which is screwed into the top of the furnace. A fusible plug which has been largely used is that adopted by the National Boiler Insurance Company and illustrated in Fig. 121. It consists of a gun-metal cap A, which is screwed into the furnace plate from the water side. Into the other end of the cap the conical plug B is screwed; this plug is in two distinct parts, the annular conical space between them being filled with an alloy of low melting-point. The inner end of the plug is provided with a narrow flange, which protects the fusible metal from the hot furnace gases under normal working conditions. In the event of the water level falling below the top of the furnace the fusible metal melts and the inner portion of the plug drops out, allowing steam to rush out. To ensure a fusible plug remaining

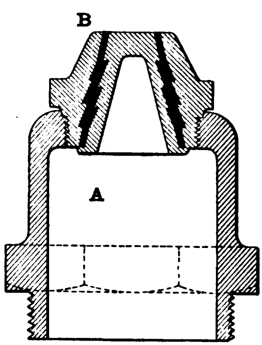


FIG. 121.—National fusible plug.

in good order, no scale or deposit should be allowed to collect on either the water or fire sides. The alloy should also be renewed at intervals of about a year, because it is found that the melting point cannot be relied upon to remain unaltered over long periods.

10. Feed-Check Valves.—To control the supply of feed water to the boiler to a steady flow of the right amount to keep the water level constant, some form of valve is required, and better results are obtained when the boiler supply is regulated by a feed-check valve on the water pipe than by the steam valve on the feed pump. The check valve should be fitted directly to the boiler within easy reach of the stoker, usually on the front end plate of the steam drum. A good design of check valve is shown in Fig. 122. The upper regulating valve A can be screwed down and the check valve B taken out for examination whilst the boiler is under steam. The flange C is bolted to the front end plate of the boiler, the pipe from the delivery of the feed pump or injector being bolted to the flange D. To admit water into the boiler, the valve A is opened by means of the handwheel, the water, under pressure from the feed pump,

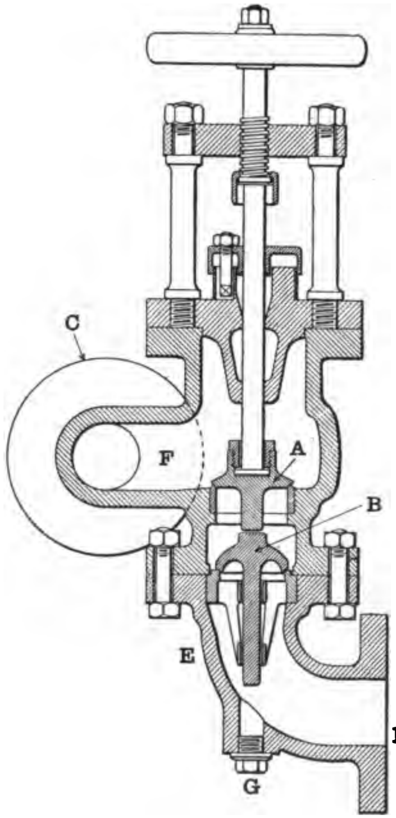


Fig. 122.—Hopkinson's accessible feed-check valve.

lifts the check valve B against the boiler pressure and so enters the boiler. The valve is a *non-return* one, for if the feed pump should fail or be intentionally stopped, the boiler pressure keeps the check valve B on its seat and so prevents any water passing through it from the boiler.

The regulating valve A is loose on its spindle, enabling it to find a perfect bed on the seat without straining the spindle. The check valve B and its seat are of solid "Platnam" metal, the seat being recessed into the body of the lower casting E and held immovably in position by the overlapping of the upper casting F. An inspection plug G is situated below the check valve. On closing the control valve on the feed main, and the regulating valve A, the elbow E containing the check valve and its seat can be removed while the boiler is under pressure, examined and replaced, sufficient water having previously been fed into the boiler to keep it at work without feed during this examination.

11. Blow-Out Cocks.—As its name shows, the blow-out cock on a boiler is simply to enable the boiler to be entirely emptied of water, and incidentally to enable a certain amount of the muddy water at the bottom of a boiler to be blown off periodically so that the mud and sediment may not continually accumulate. As already mentioned (Art. 1, Chap. VI.), it must for both objects be fitted in the lowest portion of the boiler. It is important that when closed the cock should be perfectly tight, otherwise leakage of water will be going on continually when the fires are banked, and if, as is sometimes the case, the night watchman lights the fire the lowering of the water level may escape detection. The cock should be designed to open and close easily when the boiler is under steam. Fig. 123 illustrates Hopkinson's parallel slide blow-off valve, which has many merits. It consists of two discs A with a spring between, arranged on the well-known Hopkinson Parallel Slide principle. These discs slide on seatings B, screwed into the body of the valve. Both discs and seatings are of solid "Platnam" metal, and the seatings are provided with lugs C on their inner surfaces, enabling them to be screwed out when necessary. The discs are raised and lowered into position by

means of a rack D attached to a collar embracing the discs, the rack engaging with a toothed pinion E on the end of the operating spindle F. To prevent overwinding of the spindle in

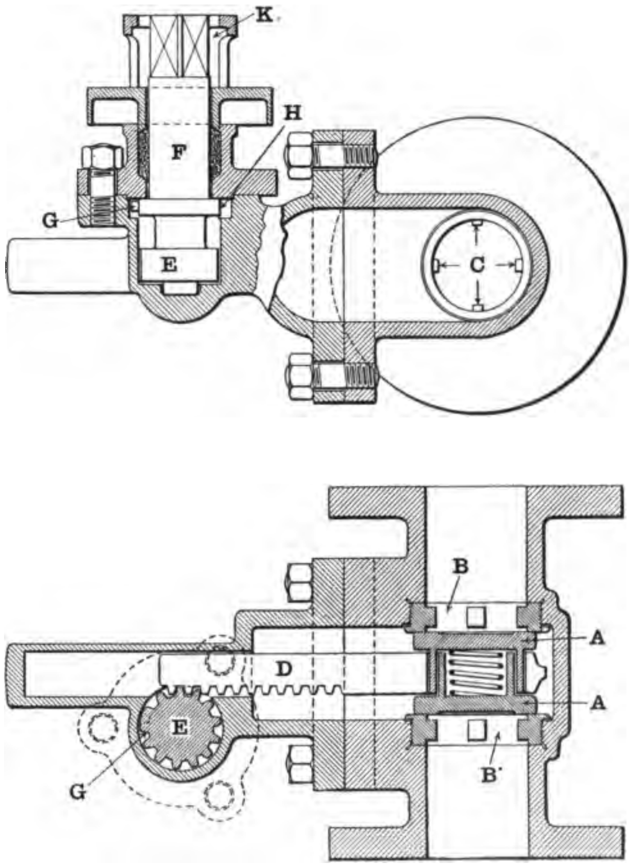


FIG. 123.—Hopkinson's patent parallel slide blow-off valve.

either direction an "Oplok" stop G is provided on the spindle, consisting of a projecting lug that comes in contact with stops H, as shown. A locking device K, on the end of the operating

spindle prevents the removal of the key when the valve is open, but not when the valve is shut. This prevents the valve from being accidentally left open at any time, and is an excellent precaution, because the boiler may be one of several working on the same steam main, and when shut down for cleaning purposes it is obvious that if the blow-off valve is left open and one of the other boilers blown off, hot water and steam may enter the idle boiler and scald any one inside it.

The valve discs A, being an easy fit inside their collar, are able to find their own bed on the seating, and the disc on the outlet side of the valve is maintained an absolutely tight fit in the closed position by the boiler pressure itself.

12. Water Gauges.—The usual practice is to fit on the front end plate of each boiler two sets of water gauges, so that the one can be checked by the other. With high steam pressures the taps should be packed with asbestos in order to render them easy to open and close. An asbestos-packed automatic water gauge by Messrs. Schäffer & Budenberg, is shown in Fig. 124. The plugs A do not bear on the metal of the casing, but upon the asbestos packing (shown black in the sectional view) contained in the four longitudinal grooves in the body of each cock. By this means leakage is entirely prevented and the taps can be easily and smoothly turned. In the event of a glass breaking, the rush of hot water automatically forces the ball B in the lower fitting on to its seat, and allows the attendant to close the tap without risk of being scalded. To keep the passages free and the gauge in working order all the taps should be opened several times during the day, otherwise the gauge might register a false water level.

In the event of a gauge breaking, in order to protect the attendant from being struck by bits of glass, some kind of transparent shield should be fitted. In the design shown in Fig. 124 the gauge glass protector is made of toughened glass moulded on wire netting, which is thus embedded in the middle of the glass, as shown in Fig. 125. The wire netting will hold the protector together even if the glass is broken. The glass protector is held in position by means of two helical springs as

shown, which will yield in case the gauge glass breaks, and thereby diminish the shock and preserve the protector glass.

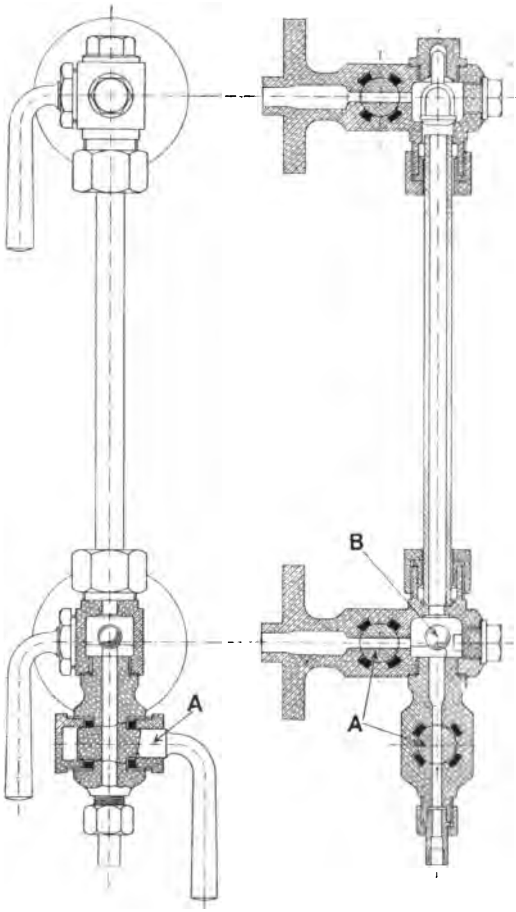


FIG. 124.—Asbestos-packed automatic water gauge.

FIG. 125.—Automatic water gauge with glass protector.

13. Pressure Gauges.—The construction of the Bourdon Pressure Gauge is shown in Fig. 126. One end of the curved

brass tube A, of oval cross section, is soldered into the boss B, while the other end is closed with the plug C, which is connected by the rod D to the toothed quadrant E. The quadrant E gears with a small toothed pinion F, to the spindle of which is fixed the indicating pointer. The steam pressure to be measured enters through the boss B into the tube A. The effect of the pressure inside the oval tube is to cause its cross section to become more nearly circular in shape. This results in the tube trying to become straighter, causing an outward movement of

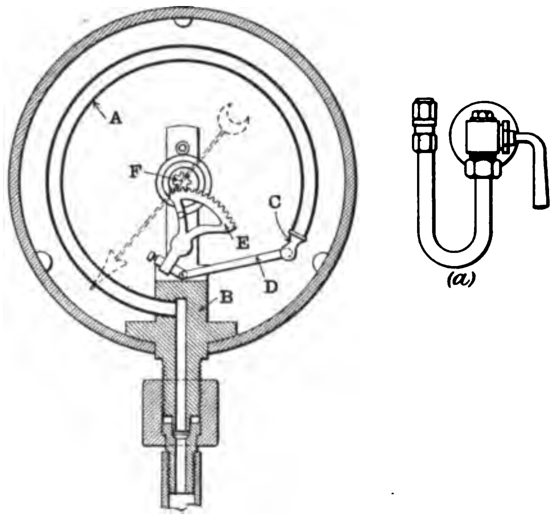


FIG. 126.—Bourdon pressure gauge.

the end C and therefore a rotation of the pinion F and a corresponding movement of the pointer over a graduated scale. Gauges of this type are not suitable for pressures exceeding 120 pounds per square inch.

For working pressures greater than 120 pounds per square inch the curved elastic tube instead of being made of brass is of steel. To protect the tube from corrosion it is sometimes tinned inside as well as outside.

The deflection of the elastic tube is dependent not only upon the pressure, but also in some degree upon the temperature, and

it is therefore important that the temperature should be kept as uniform as possible. For this reason the gauge is usually attached to a **U** bend (Fig. 126 (*a*)), the other end of which is connected to the boiler; steam condenses in this bend and forms a water seal.

14. Steam Pipes.*—To convey the steam generated in a range of boilers to the engines or other apparatus which they supply is not by any means an easy problem. Many accidents due to steam pipes bursting have occurred from time to time, and the Board of Trade and Insurance Companys' records give ample proof that steam pipes and their fittings are a constant source of trouble and danger, caused in nearly all cases by defects other than failure to withstand the working pressure. The pipes and fittings used in the average steam plant are amply strong enough to withstand the usual working pressure. For instance, a well-designed stop valve will successfully pass tests which may subject it to bursting stresses four or five times greater than any which can be set up by the normal steam pressure, and in like manner copper bends, cast-iron trees, branches, etc., have a factor of safety of from six to eight, whilst the welded steel pipe now in common use has a factor of safety of from ten to fourteen. The breakdowns that occur in practice are usually due to one or a combination of the following:—

(*a*) Water-hammer.
(*b*) Expansion or distortion due to variations in temperature.
(*c*) Want of alignment.
(*d*) Excessive temperature and vibration.
(*e*) Internal and external corrosion.

15. Water-hammer.—Water-hammer, while perhaps the easiest to guard against, is undoubtedly to blame for the majority of the more serious accidents, and the initial cause is, in most cases, faulty design supplemented by want of care on the part of those working the plant. The exact nature of the phenomenon known as water-hammer is very difficult to define,

* The author is indebted to Messrs. Alley & Maclellan for the following Arts. 14–19, from "Some Notes on Steam Pipe Design," by W. J. Poole.

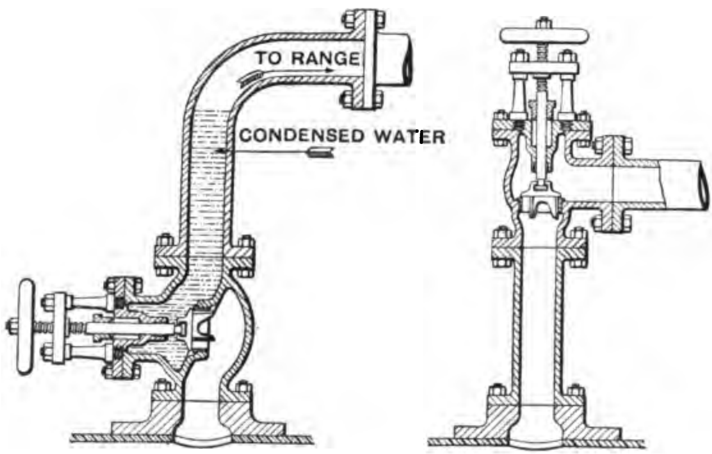
though its effects are only too well known to every engineer, while the conditions required for bringing about the trouble are simple in the extreme, *i.e.* any arrangement of the pipes or valves which makes it possible for condensed steam to accumulate in the system, or want of care in using the means provided for safely getting rid of such accumulations.

If steam is suddenly admitted to a pipe partly filled with cold water, the latter will be set in violent motion and travel along the length of the pipe in the form of waves or plugs with sufficient velocity to rupture any valve, blank flange, or other obstruction in its path. The velocity of flow and consequent disruptive effect appear to depend not only on the quantity and temperature of the water but, more than anything else, upon the rate at which the incoming steam is allowed to mix with it. For example, if steam from a boiler under, say, 150 pounds per square inch pressure is admitted to a horizontal pipe half full of water, and the steam supply carefully regulated through a small byepass valve, the probability is that, though there will be a certain amount of noise and vibration, accompanied by heavy leakage at all the joints, nothing further will happen, and if at the same time the water is allowed to escape it will do so, and by the time the steam pressure has risen to a few pounds per square inch the pipe will be thoroughly hot, free from water, and practically all leakage at the joints will have taken up. If, on the other hand, a considerable quantity of steam is suddenly admitted, say, by opening the boiler stop valve, a violent explosion is almost certain to follow, probably resulting in the destruction of the valve and serious injury to the operator.

The same result may be brought about in other ways than by the sudden admission of steam; for instance, take the case of a vertical pipe under full pressure and having a stop valve at its lower end. If this valve is left closed for any length of time, condensed steam will gradually accumulate above it as water, and if not allowed to drain away, the level of the water will gradually rise and its temperature decrease, until we have a vertical pipe containing a column of practically cold water, say under 150 pounds per square inch pressure. Now, if the small bye-pass valve which is, or ought to be, fitted to the stop valve be care-

fully opened, the water will not be disturbed, and will gradually drain away through the lower part of the pipe so that when sufficient steam has followed to equalise the pressure, the main valve may be opened with perfect safety. But if, on the other hand, the pass valve be opened suddenly to its full capacity, or any attempt is made to open the main valve, the cold water will be thrown into a state of violent commotion, intermix with the steam in the upper part of the pipe and set up water hammer of more or less severity, depending on the area of the bye-pass valve and the time taken to open it.

16. Drainage of Steam Pipes.—Many instances might be quoted to show that while pipes and valves can be, and are,



(a) *(b)*

FIG. 127.—Position of stop valve.

constructed to withstand with absolute safety pressures greatly in excess of those in common use, yet they cannot successfully resist the stresses set up by water-hammer, and that the only solution of the difficulty is to design the pipe system in such a way as to render any serious accumulation of water impossible. As a simple case take that of an ordinary boiler stop valve. If this is arranged as shown in Fig. 127 at (*a*), then whenever the

boiler is shut off from the range, water will gather in the bend and cause trouble from leaky joints and certain disaster if the valve is opened before the pipe is thoroughly drained; but by altering the arrangement to that shown at (*b*) Fig. 127, there is no possibility of water accumulating above the valve; the horizontal pipe always dips, or rather should always dip, towards the main range, and there is also no drain connection required, no leaky joints, and no risk of a momentary carelessness causing an accident. Fig. 128 shows a simple arrangement of piping in which the valves have been properly placed and the

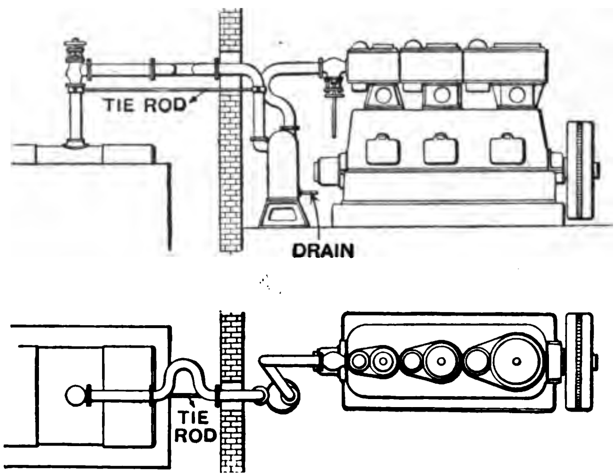


FIG. 128.—Arrangement of steam piping.

pipe arranged with due regard to drainage and expansion. It will be noted that should the boiler stop valve leak slightly, any water which may collect in the pipes will at once be carried to the separator, that only one drain is required, and that any strain due to expansion and the vibrating action which is set up in pipes having expansion bends, by the thrust of the steam pulsations, is taken care of by the tie rod. This rod also makes it possible to carry the vertical pipes on the boiler and separator up to any reasonable height without risk of straining the joints, as would happen if they absorbed the thrust of the

expansion bend acting on the unsupported ends of the vertical pipes.

While the question of locating drains on high-pressure steam pipes is one of considerable importance, the system of pipes leading from the drain valves, or traps, to the hot well is also worthy of attention. In theory every drain valve or trap should have its discharge outlet in close proximity to the apparatus with which it is connected, and in clear view of the attendant. While this is quite possible in some cases it is out of the question in others, the alternative being to instal a complete system of pipes between the trap outlets and the hot well. If it were not for the fact that these pipes are often of very considerable length, and therefore costly to instal and keep in order, the best method would be to have a separate and independent pipe from each trap. It would then be possible to locate defective valves, and to obviate the risk of blowing back into boilers or separators not under pressure, but in most cases the expense and complication is prohibitive.

Another method is to run a length of ordinary cast-iron water pipe about 3 inches diameter along the division wall between the engine and boiler house, support the pipe on any convenient form of bracket, and connect the main stop valve or other drains to the cast-iron pipe by means of short lengths of $\frac{3}{4}$-inch copper tube and screwed nipples. The cast-iron pipe should have a good fall towards the hot well, and no dips or pockets where condensed water could gather; the higher end should be left open, but be led outside the building, and the amount of vapour coming from it will be a fair indication of how the traps are working, any excessive discharge of steam being noticed at once. The traps themselves should be placed on wall brackets and kept as close to the steam pipes as possible.

17. Protection against Expansion and Vibration.—The alteration in length of steam pipes due to changes of temperature is quite appreciable, particularly in long lengths of pipe. In moderate lengths of pipe the necessary bends (if there are any) will probably of themselves make the range sufficiently elastic, but if the range of pipes is long,

a definite expansion device must be added. A very usual plan is to introduce a copper bend, but even at moderate temperatures, say between 400° and 500° F., copper and gun-metal begin to give trouble, and expansion bends constructed of copper are not to be trusted for any length of time even when frequently annealed. When superheated steam is used, copper pipes and bends are therefore prohibited, but mild steel bends—provided that the radius is reasonably large and care is taken to provide extra heavy flanges, preferably welded on—are perfectly satisfactory and possess a considerable amount of elasticity. An 8-inch pipe bent into the form shown in Fig. 129

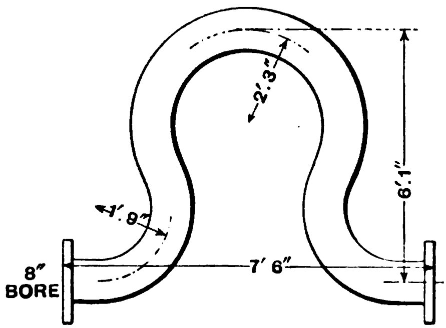


FIG. 129.—Steel expansion bend.

can be compressed about $\frac{3}{8}$ of an inch when cold, with a pressure of 450 pounds applied at the ends, and $\frac{13}{16}$ of an inch with a pressure of 900 pounds. Such a bend should, of course, be kept horizontal and the outer ends of the pipes connected with it should be firmly secured.

The use of superheated steam has brought about conditions which, as far as steam pipes and valves are concerned, preclude the use of materials liable to deterioration under high temperatures. In most water-tube boilers fitted with integral superheaters (Chap. VII.) no provision is made for bye-passing the gases or otherwise controlling the temperature, the latter varying with the load on the boiler. This is particularly the case in electric lighting plants during "peak" loads and

abnormal conditions due to fog and like causes. If then, occasional overloads have to be dealt with, and the furnaces are forced to meet the extra demand for steam, the temperature in the superheaters may rise to a point very much higher than is usual under normal working conditions; the steam temperature through the range of pipes is correspondingly high, and though the time during which the pipes and valves are exposed to these conditions may be comparatively short, perhaps not more than a few minutes daily, yet the crystallizing or "shortening" effect on copper pipes and gun-metal or cast-iron fittings will be none the less certain. In such cases the choice of materials is for all practical purposes restricted to steel in one form or another. Again, when steam turbines are installed it is usual for economic reasons to supply them with steam at a temperature of from 470° to 550° F. so that if the steam range is of average size, the temperature at the superheater stop valves will not be less than 600°, probably nearer 700° F. As the pipes and fittings will be continuously exposed to this temperature, the suitability of the materials employed in their construction becomes a question of serious importance.

When a number of engines of small or moderate size are connected to the same pipe system and stand on the same foundation, or at least in the same building, it is sometimes difficult to prevent the pipes from vibrating and at the same time to ensure the necessary freedom for expansion and contraction due to temperature changes. The modern high-speed two-crank compound engine usually has cranks set at 180 degrees apart, and whether single or double-acting has a tendency to rock in a direction parallel to the crank shaft. An engine of this type usually has the steam separator bolted to one end of the bedplate, with the result that the rock or swing of the latter is transmitted in an aggravated form to the nearest part of the steam range, and when several engines swing synchronously, the pipes, if not properly supported, may vibrate to a dangerous extent. The pipes should therefore be arranged in such a way that whilst they are free to move in a direction parallel to their length (in order to provide for expansion), movement in other directions should be restricted as far as

possible. Fig. 130 shows a main steam pipe carried in cast-iron frames A and rigidly anchored to the wall of the building at a point midway between the ends of the pipe. The frames are fitted with adjustable rollers bearing on the top as well as taking the weight of the pipe, and the anchor plate or bracket is provided with heavy clamping straps and fitting strips. The bracket rollers may be placed at the sides of the pipe in cases when the thrust from a horizontal branch connection has to be provided for. The extra first cost of such an arrangement as compared with ordinary slings is small, and it does away with such objectionable makeshifts as wooden wedges and the

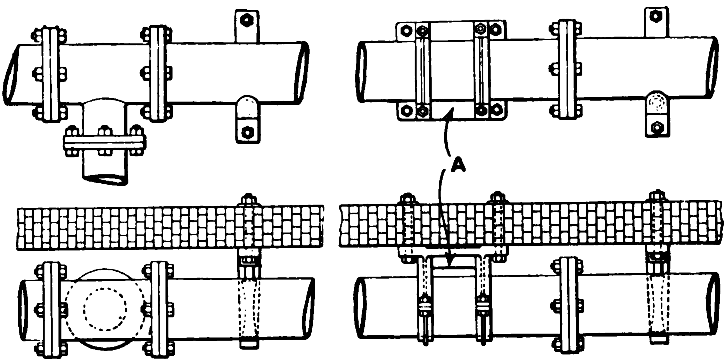


FIG. 130.—Method of supporting steam pipes.

necessity for building brickwork close round the pipes where they pass through the walls, whilst the expenditure on general repairs, such as the remaking of joints and the renewal of pipe-covering materials, is likely to be greatly diminished.

18. Erection and Cleaning.—It frequently happens that although all the foregoing considerations, *re* drainage, expansion, excessive temperature, etc., receive careful attention, the actual installation of the pipes is carried out in a more or less negligent manner, one reason being that work of this kind is usually left until the last and therefore probably done hurriedly. Most steel pipes and castings as received from the makers contain a certain amount of mill scale and foundry sand, and if this

is not removed and the pipes thoroughly cleaned before being placed in position, the probability is that the stop-valve seatings will be badly damaged and have to be replaced several times before they will remain in good working order. This trouble with valves, more especially with parallel slide valves, could be easily avoided if live steam were blown through the pipes for a few minutes before the plant was started and before any attempt was made to close the valves; the engine stop valves would of course be kept closed and temporary connections made in order to carry the steam out of the engine room. In the same way, much trouble with steam traps would be obviated if care were taken not to connect the traps to the separate drains until the latter have been in use for a few days; otherwise the trap valves have to pass all the grit, etc., intercepted by the separators, and there is always some, no matter how well the pipes have been cleaned.

Want of alignment sometimes causes trouble by throwing excessive strains on the flanges of stop valves, etc. This is brought about, as a rule, by the flanges having been forced into contact with each other by means of the jointing bolts, instead of fitting into place as they should do. The flanges of modern steel pipes and valves are usually of ample thickness, and if they do not come fairly together in the first instance, they should be taken down and refaced, a thin ring of sheet steel being put in to make up the length if necessary.

When erecting heavy pipes, every length should be placed in position and properly supported and levelled by its own slings or brackets, when it will usually be found that several lengths have to be altered before the flange faces come into alignment, and not until this has been done and every pair of flanges inspected by some responsible person, should the various lengths be bolted together permanently.

19. Internal and External Corrosion.—As already mentioned, steel is now very largely used in the construction of all classes of steam, exhaust, and feed pipes, but it is, as a rule, only in the latter that serious trouble is caused by internal corrosion. If the feed water contains lime salts, the latter

become deposited in the economiser and feed connections and more or less effectually protect the pipes from corrosion, but if the water is very free from lime, and if at the same time air is introduced by the feed pump or otherwise, internal pitting will be set up, and probably do considerable damage before it is discovered and steps taken to prevent further mischief.

With some feed waters it is practically impossible to use steel pipes, as they have to be renewed so frequently that copper ones are much cheaper in the end, and will usually last as long as the boilers. But even with condensed water every care should be taken to prevent, as far as possible, the introduction of air by the feed pump. The pump should be of the slow-running type, in order to keep the water running steadily with a low velocity. The feed water should flow into the suction pipe under a head of from 6 to 8 feet, and all water glands should be kept well packed, and all snifting valves and pet cocks removed from the valve boxes. If the feed pipes rise above the level of the boiler check valves, an air vessel should be placed at the highest point, and a small drip pipe should be taken from the air vessel back to the hot well; this will prevent air gathering in the higher parts of the pipe system. If air does gather there, trouble will arise not only from corrosion and pitting, but from objectionable vibration and hammering of both pipes and check valves.

The use of separate water and air pumps in connection with surface condensers will do much to obviate trouble from corrosion in feed pipes, for when the air is removed by a dry air pump, and the water pumped out separately, the latter is much more suitable for boiler-feeding purposes than it is after being charged with air in a wet-air pump.

External corrosion does not, as a rule, give much trouble, but under certain conditions the combined action of heat and moisture on asbestos pipe covering will set up pitting. This, however, can be prevented by painting the pipes with any good graphite paint before the covering is applied.

20. Lagging of Steam Pipes.—In order to reduce the radiation losses from steam pipes, it is essential to lag them with a good heat insulator. It does not necessarily follow,

however, that the best material to use for this purpose is the worst conductor of heat; other considerations, such as mechanical strength, working temperature, resistance to moisture, etc., should also be taken into account.

Several important contributions have been made in recent years, to our knowledge, of the problem of heat insulation.* The important conclusions arrived at by these investigators may be briefly summed up as follows :—

Dr. Nusselt found that the heat conductivity of the lagging materials usually used in practice was not a constant quantity, but increased with the temperature, as shown in the following table :—

Material.	Temperature, ° C.							Density.
	50	100	200	300	400	500	600	
Cotton waste	0·054	0·059	—	—	—	—	—	0·081
Silk cord	0·047	0·052	—	—	—	—	—	0·147
Infusorial earth (loose) .	0·060	0·066	0·074	0·078	—	—	—	0·350
,, ,, (baked)	0·071	0·078	0·092	0·106	0·120	—	—	0·200
Compound	0·070	0·076	0·081	—	—	—	—	0·405
Asbestos	0·153	0·167	0·180	0·186	0·192	0·198	0·204	0·576

Kieselguhr or Infusorial Earth.—The conductivity rose steadily with the temperature. The solid paste did not insulate so well as loose dry powder, especially at temperatures of 300° to 400° C.; perfect dryness is therefore essential.

Compound consisting of Cork, Asbestos, Infusorial Earth, and a Cement.—In practice these ingredients are mixed with water to a thin paste and painted on steam pipes, each layer being allowed to dry before a new layer is applied; cloth is finally wrapped round the compound and coated with oil paint. In this case also the insulating power was impaired by the addition of water.

* See the following papers: *Engineering*, Jan. 1, 1909, Dr. Wilhelm Nusselt; *Engineering*, July 9, 1909, Mr. Charles R. Darling; and "Testing of Heat-Insulating Materials," by Frederic Bacon, M.A., Section G of the British Association, Sheffield, Sept., 1910.

Asbestos.—This was not found to be a really good insulator, but it has the advantage of withstanding high temperatures.

The relatively high thermal conductivity of asbestos which Dr. Nusselt's investigations bring out, and its inferiority to cork, cotton, and silk, may partly be due to the more compact character of the asbestos. Further experiments on the effect of close packing would have been desirable.

Mr. Bacon's results refer mainly to woods, corks, slag wool, and other materials primarily intended for insulating the walls of cold stores. He comes to the conclusion that it is advisable to pack *silicate cotton* as firmly as possible, especially as this ensures that the material will not afterwards subside and leave holes which cannot afterwards be filled up. The results for asbestos lead one to the conclusion that its value as a heat-insulator is due rather to its fire-proof qualities than to its low conductivity, which appears to be no lower than the conductivity of wood.

In Mr. Darling's research, all the materials tested were intended as steam pipe or boiler coverings, and the outer surfaces were exposed to convection and radiation so as to imitate practical conditions as far as possible. All the samples were well dried before being tested, and the apparatus used consisted of a closed metal vessel containing an electric heater and lagged with the material to be tested, tests being made with temperatures varying from 250° to 500° F. His results may be briefly summed up as follows:—

Magnesia Lagging (85 per cent. of light magnesium carbonate and 15 per cent. of white asbestos fibre). The insulating power improves with increase of temperature, probably due to the expulsion of moisture as the temperature rises. He says, "Blue asbestos, when used as fibre, resembles magnesia in this respect, and also, according to some observers, silicate cotton. Practical experience in the use of these materials for high temperature work confirms these observations, which explain why the superiority of some laggings over others is more manifest at high temperatures."

Plastic Composition (containing vegetable fibre and several other ingredients, and intended to be applied with a trowel).

The insulating power is not so good as magnesia, and shows a falling off as the temperature rises. Owing to the great difference in the densities of what purported to be the same laggings, he found himself unable to give comparative figures for a number of specific compositions.

Effect of Density.—"Inasmuch as a porous solid is a better insulator than either an air-space or the same solid in a non-porous form, it follows that in all cases a certain ratio of air-space to solid must exist at which the insulating power is a maximum. Many laggings do not permit of wide variations being made in the density, but wherever possible it would be a distinct advantage to determine the best density and to make up the lagging accordingly. A striking example of the effect of density is to be found in the case of kieselguhr, which in the native state is one of the best mineral insulators. When injudiciously blended with binding material, the porous character of the kieselguhr is largely destroyed, and the resulting lagging is relatively an inferior insulator. The correct manufacture of a lagging is of equal importance as its composition."

"Engineers are largely guided by custom with regard to the thickness of lagging, and whilst in most cases the thicknesses adopted are fairly suitable, there are many instances in which hot surfaces are absurdly overlagged or underlagged, owing to the absence of data from which the correct thickness could be determined." Mr. Darling shows in his paper how the most economical thickness to use for a given purpose can be deduced, and further states that "it will generally be found that, for high temperature work, the superior insulators, in spite of their greater prime cost, are cheaper in the end than inferior types; whilst at low temperatures the contrary may prove true. Each case must evidently be decided on its own merits, the factors to be considered being prime cost, working temperature, and estimated duration."

CHAPTER X

THE SELECTION AND TREATMENT OF A BOILER FEED WATER

1. The Selection of a Boiler Feed Water.*—In order to decide whether or not a certain water will prove suitable as a boiler feed supply, experience and care are necessary. With a water containing considerable amounts of scale-forming impurities, or containing any corrosive acid, it is evident that scale, or corrosion, or both will result. In waters having small amounts of impurities the solution of the problem is not easy, because so many different substances may make up these impurities, each affecting the other to a greater or lesser extent, depending on the relative amounts of the substances present. The type of boiler and the conditions under which it works also enter into the problem.

It is not always safe to base an opinion upon the number of grains of impurities per gallon in a feed water, for in some waters as much as eight grains may be present without any scale resulting, except possibly in the feed-water heater and pipes, while, on the other hand, many waters with only three or four grains per gallon cause hard scale, and in some instances corrosion.

In the generation of steam the heat alone causes the precipitation of some of the impurities in the feed water; it also drives off certain gases and volatile substances, and brings about the decomposition of some of the soluble salts. The heat, together with the concentration resulting from evaporation,

* The author is indebted to Mr. J. C. W. Greth's paper on "The selection of a boiler feed water," read before the *Engineers' Society of Western Pennsylvania* in 1910 for much of the matter given in this article, and for the analyses of water.

further produces reactions between the soluble substances, causing the precipitation of scale-forming matter and the liberation of corrosive acids. The suspended impurities, while a factor to be considered, are of minor importance, and the obvious remedy is their removal by sedimentation or filtration. The soluble impurities carried by a water may be divided into two classes, *organic* and *inorganic*. The organic matter is usually small in amount, and seldom plays much part in scaling or corroding boilers.

The inorganic impurities, however, are the usual cause of most boiler troubles. The most common soluble inorganic impurities are lime, magnesia, iron, and sodium, in combination with carbonic, sulphuric, hydrochloric, and in some cases nitric acid. There are also found a small amount of silica, oxides of iron and aluminium, some free carbonic acid, and in some waters free sulphuric acid, in others hydrogen sulphide.

These soluble mineral substances may be divided into two classes, *non-scale forming* and *scale forming*. The non-scale forming impurities consist of the sodium salts, which are soluble under all conditions in the boiler, and are not objectionable in feed water unless present in very large quantities, for they are both non-scaling and non-corroding. The sodium salts, however, are often charged with being the cause of priming and foaming; it is a common belief among engineers that these two evils are directly caused by sodium salts, of which sodium carbonate (soda ash) is the one most feared. When used in correct amounts for water purification, soda ash is not present in the treated water as sodium carbonate, but as sodium sulphate, chloride, or nitrate.

The priming and foaming in boilers which sometimes occur immediately after beginning the use of a water softened and purified with lime and soda ash or caustic soda has led to the belief that the soda treatment was the sole cause of it. But close observation and many analyses of concentrated boiler water now warrant the belief that the cause is to be found among the following factors: the nature of the impurities in the old scale, or those introduced into the feed water by the exhaust steam; the design of boiler; the water space; the

steam space; the arrangement and size of steam piping; irregularity of load, and improper firing and feeding.

That foaming is not directly chargeable to the sodium salts is almost conclusively proved by the fact that with a properly treated and purified water in which the sodium sulphate, chloride, nitrate, and carbonate total 100 grains per gallon, with sludge-forming substances less than 3 grains per gallon, and with no scale-forming substances present, no foaming results if the boilers are clean. But if such a water be fed into a boiler badly scaled, the scale begins to disintegrate, and during the time the old scale is coming off, foaming often occurs. Experience has shown that priming and foaming cease as soon as the boiler becomes clean, if the boiler is regularly blown-off and occasionally emptied and refilled with fresh water.

The amount of sodium salt that can be carried by a water without detriment to its successful use for boiler feed is greater in those waters that are practically free from scale-forming salts, suspended matter, and oily matter. The concentration of the sodium salts can be controlled by proper blowing off. Being able to operate boilers without scale and corrosion more than compensates for any water at boiler temperature that it may be necessary to waste by blowing off in order to control concentration.

Temporary and Permanent Hardness.—The soluble scale-forming substances are usually divided into two classes, those which cause *temporary* hardness, and those which cause *permanent* hardness. Temporary hardness is due to the carbonates of lime, magnesium, and iron; the precipitation of these begin at temperatures below 212° F., and is practically completed by continued boiling at this temperature. Permanent hardness is due to the sulphates, chlorides, and nitrates of lime, magnesium, and iron, which, as a rule, are not precipitated at temperatures below 212° F.

It has been shown that the amount of impurity which a water contains is not always a criterion of its adaptability as a boiler water. The evaporation of water from the boiler results in a continual concentration of the impurities introduced with the feed water. Consequently there is a constant increase in

the amount of impurities in the boiler, the results of which are apparent as suspended matter, scale, corrosion, and increased density. Scale and corrosion are closely related because of the number of salts which, as a result of heat and concentration, may either decompose or react, liberating acids and forming solids which are precipitated directly or after concentration. The precipitated solids form scale, and the liberated acids cause corrosion.

Scale can nearly always be attributed to the lime and magnesia salts in solution in the water. The scale deposited in exhaust steam feed-heaters (where the temperature is below 212° F.) is almost invariably that of the temporary hardness. Heating the water drives off carbonic acid (CO_2) and converts the bicarbonates of lime and magnesia into the practically insoluble carbonates which are deposited as sludge or scale. Those waters which possess considerable temporary hardness quickly reduce the efficiency of the heater and necessitate frequent cleaning. The precipitation of the carbonates is not an instantaneous action, and therefore the pipe lines from the heaters to the feed pumps, and from the heaters to the boilers eventually become clogged with scale.

To decide on the suitability of a water for boiler feed an accurate analysis of the water is essential. This analysis should be complete, and be carefully made by a chemist who makes a speciality of water analyses, and not by one who is unfamiliar with the characteristics of various waters in relation to boiler operation. The analysis of scale is of very little value in determining the character of the water from which the scale was deposited, since, as already pointed out, the original character of the impurities in a water are not always apparent in the scale, which results from their reactions and concentration.

Neither does the amount of scale-forming impurity in the water always indicate the kind or amount of scale formed, but an analysis of the water *will* show clearly whether or not scale and corrosion will result. Such general statements as that waters containing principally the carbonates of lime and magnesia will form a comparatively soft scale, and that the calcium

ANALYSES OF FEED WATERS BEFORE AND AFTER CONCENTRATION IN THE BOILER.

Grains per U.S. Gallon.

	A Well Water.	B Concentrated Boiler Water.	C Well Water.	D Concentrated Boiler Water.	E Feed Water.	F Concentrated Boiler Water.	G Well Water.	H Concentrated Boiler Water.	I Lake Water.	J Concentrated Boiler Water.
Volatile and organic matter	1·30	4·20	0·65	2·70	0·35	0·65	0·65	3·80	0·35	2·90
Silica	0·55	0·55	0·25	1·25	0·35	1·20	1·25	11·40	0·35	0·45
Iron and aluminium oxides	0·15	trace	0·35	0·65	0·20	0·15	trace	0·15	0·10	trace
Calcium carbonate . . .	3·25	1·00	0·27	1·25	1·00	1·25	1·00	1·50	1·02	2·02
Calcium sulphate	13·55	78·80	1·67	11·42	0·58	3·71	0·68	8·70	2·14	7·68
Calcium chloride	—	—	—	—	—	—	none	7·53	none	1·19
Calcium nitrate	—	—	—	—	—	—	none	22·78	none	0·33
Magnesium sulphate . .	4·11	6·36	0·15	0·87	0·21	0·12	—	—	0·93	none
Magnesium chloride . .	—	—	0·27	none	0·11	none	0·53	0·85	0·32	none
Magnesium nitrate . . .	—	—	—	—	—	—	0·48	none	none	0·32
Sodium chloride	0·99	10·55	0·13	2·64	0·16	0·46	1·32	66·89	0·10	12·77
Sodium sulphate	0·28	19·81	none	4·62	none	0·60	—	—	—	—
Total solids	24·18	121·17	3·74	25·40	2·96	8·14	5·91	128·60	5·31	27·66
Suspended matter . . .	1·25	200·40	0·60	13·30	3·15	2·50	0·15	2·95	0·05	0·80
Free carbonic acid . . .	1·43	none	0·22	none	0·22	none	1·32	none	0·33	none
Incrusting solids	21·61	86·61	2·96	15·44	2·45	6·43	3·94	52·91	4·86	11·99
Non-incrusting solids . .	1·27	30·36	0·18	7·26	0·16	1·06	1·32	66·89	0·10	19·77

sulphate will form a hard scale, and further, that it will increase the hardness of the carbonate scale, should be made with caution, for there are many instances where a hard scale has been formed from waters containing mainly the carbonates of lime and magnesia, and also where the scale is quite soft in the presence of a considerable amount of calcium sulphate. The nature and amount of scale formed in a boiler depends largely on the rate at which the boiler works; the type of boiler also has a bearing on the hardness of the scale.

Of the various scale-forming impurities in boiler feed water, probably none causes more trouble than calcium sulphate, which seems to be in nearly all cases the cement which holds and binds the other precipitated substances into scale. Calcium sulphate is usually considered insoluble at a temperature of about 300° F. This may be true in distilled water, but its solubility is raised by the presence of other substances even at temperatures far above 300° F. This is illustrated by the analyses A and B in the accompanying table. A is the analysis of a well water used for boiler feed, B is the analysis of the concentrated water in the boiler which worked at a gauge pressure of 125 pounds per square inch (temperature 352° F.).

Analysis C is that of a well water, which would be considered by many an ideal natural water for boiler feed. This water has a total of 2·96 grains of incrusting solids, but the character of these solids is such that both scale and corrosion result from its use. Analysis D shows this water after concentration in the boiler, indicating that reactions have taken place, and the cause of scale formation, as well as of corrosion.

A water containing in solution calcium carbonate and magnesium sulphate is quite a common combination of scale-forming impurities, and when these are present in small quantities they are not generally considered objectionable, as the calcium carbonate is supposed to form only soft scale or sludge, while the magnesium sulphate when present alone is not supposed to form scale; but these two substances will react under concentration at the temperature in the boiler forming calcium sulphate and magnesium carbonate. We thus

get the most objectionable of all scale-forming salts, calcium sulphate. Analyses E and F show a water of this character before and after concentration.

The reaction of the chloride and nitrate of magnesium with calcium carbonate are also quite common, resulting in the formation of scale, and also causing corrosion both above and below the water-line. Analyses G and H show a water of this character before and after concentration.

From the above it is evident that a water to be good for boiler purposes must not contain either calcium sulphate or magnesium sulphate. Magnesium sulphate is also objectionable for another reason, for in the presence of sodium chloride, which is found in practically all waters, it may, after concentration at the boiler temperature, react with the sodium chloride, forming sodium sulphate and magnesium chloride; the latter then dissociates into magnesium hydrate which is precipitated as sludge or soft scale, and into hydrochloric acid, which corrodes the boiler. This is probably one of the reactions indicated by the analyses C and D.

Magnesium chloride is always objectionable because it is easily dissociated into magnesium hydrate and hydrochloric acid. In waters containing much calcium carbonate, the hydrochloric acid will form the calcium chloride and liberate carbonic acid. The nascent carbonic acid thus liberated is corrosive, and the corrosion is increased by the presence in the water of oxygen absorbed from the air. The dissociation of calcium chloride after concentration into calcium hydrate and hydrochloric acid is indicated in the analyses I and J.

Magnesium carbonate is the least objectionable of the magnesium salts found in natural waters, yet it is frequently the cause of corrosion due to the liberation of carbonic acid. It would seem hardly necessary to mention that any free acid is objectionable, yet in many boiler plants acid waters are used, and every known make-shift is employed to overcome the deleterious action of the acids.

From what has been said above it will be evident to the reader that to obtain economy in the operation and maintenance of steam boilers, a water supply should not contain any

appreciable amount of suspended matter or acids, and should be free from the sulphates, chlorides, and nitrates of lime, magnesium, and iron. The carbonates of lime and magnesium should be as low as possible for the best results, to prevent deposits in feed-water heaters, and to keep the sludge in the boilers at a minimum. With a proper boiler feed water it should be possible to operate boilers without expensive repairs year in and year out without scale or corrosion in heaters, pipe lines, or boilers. Natural supplies of water that give such results are rare, hence some means must be employed to prevent scale and corrosion. That the problem of overcoming scale and corrosion is one that is forced upon the steam user is evident from the number of boiler nostrums and mechanical devices on the market for purifying the water, either in the heaters or in auxiliary apparatus attached to the boilers. None of these methods even approximate to the results that are obtained with natural, soft, clear, non-corrosive water, but by properly softening and purifying the water before feeding it into the heater and boiler, the working obtained with natural, soft clear water can be practically equalled. The proper softening and purifying of water will eliminate or neutralise any acid, will completely remove all permanent hardness, and will reduce the remaining scale-forming substance to less than three grains per gallon. Any method that fails to produce these results is either faulty in the design and operation of the apparatus, or wrong in theory or application or both.

2. Corrosion.—As mentioned in the previous Article, corrosion may result from the concentration of the boiler water and the accompanying reactions which take place between the impurities. This, however, is not the only cause of corrosion, for many waters, although they are apparently very pure, are at the same time corrosive. In such cases the causes of corrosion are doubtless due to the presence in the water of very small quantities of carbonic acid and oxygen. The carbonic acid attacks iron, forming carbonate of iron, which, in the presence of water containing oxygen, is converted into iron oxide; carbonic acid is then liberated and free to re-attack the iron. Thus a very small trace of CO_2 may cause much corrosion.

It is almost impossible, even under laboratory conditions, to completely remove all this gas from the water. In condensing plants, however, a great proportion of the carbonic acid and oxygen is removed, since the boiler feed water consists of condensed steam which should be practically free from air. Even in these cases, however, air is often introduced into the water in large quantities, because quick-running feed pumps will not work smoothly without air (Art. 1, Chap. VIII.). In order to prevent this admission of air into the condensed feed water, it is necessary to use slow-running pumps which do not require air for their successful working; it should be pointed out, however, that all natural waters contain a fairly large quantity of air in solution, and there would be very little, if any, advantage in using slow-speed in preference to high-speed pumps for such water.

Although traces of carbonic acid found in almost all waters cannot be entirely removed, they can be neutralised by the addition of caustic lime or caustic soda, but in order to obtain satisfactory results they should be applied continuously, not in occasional doses.

The influence of other chemicals on the corrosion of iron and steel in the presence of *cold* water has from time to time been published by various authorities. Prof. Heyn has found, for instance, that chromate of potash, permanganate of potash, and arsenic acid (all of which are very powerful oxidisers) are most beneficial in reducing corrosion, probably on account of the formation of a thin protective film on the iron or steel plates. Very little information is available on the behaviour of these substances at high temperatures, and it would be unsafe to say what effect they would have in steam boilers.

It is also found that weak sulphuric acid has a stronger corrosive effect on strained mild steel than on unstrained steel, but the difference is not large enough to allow this test to be used for showing which of several pieces of steel have been strained the most. It is possible that the difference may be sufficiently great to explain in some degree the cause of grooving which frequently occurs in boilers of the Lancashire type. Those portions of a boiler which are the most severely strained

may be more severely attacked by the salts in the boiler water than other portions, and as the metal plate in these regions get thinner, the local stress will increase, and with it the corrosive action, particularly if the reduction in thickness occurs in thin lines or grooves. If this is so, then it would follow that grooving, as distinct from pitting, would proceed more rapidly in those boilers that contained large quantities of dissolved salts.* It is difficult to say with certainty whether this occurs in practice or not; many boiler experts agree that it does.

3. Sea Water.—Sea water varies in composition to a considerable extent, but the following analysis may be taken as giving about the average quantities of the various constituents.

Calcium carbonate . . .	9·20	grains per gallon.
Calcium sulphate . . .	112·45	,, ,, ,,
Magnesium sulphate . .	160·26	,, ,, ,,
Magnesium chloride . .	261·91	,, ,, ,,
Sodium chloride	1870·86	,, ,, ,,
Total solids in solution . .	2414·68	grains per gallon.

The remarks made in the previous article apply with even greater force to sea water because of the very much greater quantity of salts present. Under certain conditions, sea water in immediate contact with heated iron or steel surfaces becomes acid, by the conversion of the magnesium chloride into hydrochloric acid and magnesium hydrate. The hydrochloric acid dissolves a certain quantity of iron from the boiler surfaces, forming chloride of iron; as soon as the chloride of iron is formed it is decomposed by the magnesium hydrate already liberated, precipitating oxide of iron and re-forming magnesium chloride. In other words, the acid formed in the boiler is merely developed locally whilst the water is in contact with the hot tube, and is immediately afterwards destroyed by reuniting with the magnesium hydrate. Hence iron is never found in solution in the boiler water, and the water being evaporated in the boiler itself would, if no acid were introduced

* C. E. Stromeyer, *Memorandum Manchester Steam Users' Association*, 1909.

with the feed water, never be found acid when tested. The oxide of iron deposited is ferrous oxide and is black and remains so unless air is allowed to get into the boiler, in which case it becomes ferric oxide and changes in colour from black to brown or red.

If corrosion is to be prevented, sea water must be kept out of boilers, and this can only be done by making and keeping the condensers tight; failing this, all type of marine boilers will be always liable to corrosion.

Every ton of sea water of the average composition given above allowed to enter the boiler means the introduction of about 75 pounds of solids, and to render such water non-corrosive about 8 pounds of quicklime (CaO) or 45 pounds of soda would be required per ton, quantities which are prohibitive when there is any considerable leakage of sea water into the feed. The use of lime has the further disadvantage that out of the 8 pounds mentioned above, 5¾ pounds would be deposited as scale. Sea water itself has sufficient calcium sulphate to produce 3½ pounds of scale for every ton, so that if the proper proportion of lime be added, the total quantity of scale which will be deposited is 9¼ pounds. Although this scale, when evenly coated over the boiler surfaces, may protect them from the corrosive action of the magnesium chloride, it is at the best only an expensive and unsatisfactory remedy, since it increases the consumption of fuel, and, when accumulated, may damage the boiler by overheating.

The alkalinity of boiled sea water is a false alkalinity, in that it does not prove the non-corrosiveness of sea-water or the absence of corrosive elements, and is due to a re-solution of previously precipitated magnesia. This alkalinity is very slight, indeed, and no true alkalinity can be given to sea water until more than 45 pounds of soda crystals have been added to each ton of sea water. Any addition of soda in excess of 45 pounds will give a true alkalinity, which will, of course, be destroyed by a further leakage of sea water.

Evaporators are essential for making up the loss of water due to leakage from the glands and joints, but they must be blown down before the brine becomes too concentrated;

otherwise the magnesium chloride will be decomposed and give off hydrochloric acid, which will pass over into the boilers with the distilled water and thus render the make-up feed water acid. The action of the acid formed in this way differs from that formed in the boiler by the decomposition of sea water, inasmuch as it does not immediately afterwards become destroyed by reuniting with the magnesia, but is carried in with the fresh make-up feed and is held in solution throughout the entire volume of boiler water. It is thus in a position to attack all parts not protected by scale or otherwise.

Experience points to the use of lime as being the more generally satisfactory on a voyage, while soda may be used in cases where the boiler water is acid through the density of the brine in the evaporator being excessive, and where no vegetable oil has been allowed to enter the boilers, and on entering port at the end of the voyage.

A small amount of salt water is sure to get into the boilers, even under the most favourable conditions—through priming of the evaporator, or a slight leakage from the condenser—and it is an excellent plan to continually use a small quantity of milk of lime to neutralise it. One pound of lime per 1000 indicated horsepower dissolved in fresh water and fed in every day may suffice. The lime used is the ordinary unslaked lime of commerce, and it should be finely powdered and kept in a dry place—for instance, on the stokehold gratings. Milk of lime is a mixture of about one pound of lime to a gallon of water, and this should be strained through wire gauze before use in order to remove any lumps or solid impurities.

When starting with new boilers on a voyage for the first time, five pounds of lime should be put into the boilers for every 1000 horsepower (this lime should be dissolved in water, strained, and put in through the manhole), and two pounds of lime per day for every 1000 horsepower should be passed through the hot well, as milk of lime, for about six days, and for the remainder of the voyage about one pound per day per 1000 indicated horsepower. At the end of the voyage the boilers should be examined to see if they have a thin coating of lime scale on their internal surfaces. If this is not the case

and the water shows an improper colour, black or red, the use of lime should be discontinued.

The boiler water should be tested daily, and if it is found to contain more than about 100 grains of chlorine per gallon, the amount of lime used should be increased. If the boiler water at any time be found acid, a solution of carbonate of soda should be added to the feed at the rate of a bucket of soda solution per hour until the water just turns red litmus paper blue, after which daily additions of soda or lime should suffice to keep the water in a safe alkaline state. Carbonate of soda is effective in changing sulphate of lime into sulphate of soda, which is soluble and harmless. Carbonate of lime, which is also formed, may be easily blown or washed out.

In all cases, on entering port, soda crystals dissolved in fresh water should be added to the feed, as this will tend to soften the scale and render the boilers more easily cleaned. The use of soda *at sea* in boilers into which vegetable oil has been allowed to enter, is sometimes attended by trouble on account of the soapy scum which forms on the surface of the water being carried by priming into the high-pressure cylinder, and in such cases lime alone should be used.

4. Estimation of Hardness.—The hardness of a sample of water is determined by finding how much of a standard soap solution is required to produce a permanent lather with a known volume of the water. This gives the *total* hardness, *i.e.* the temporary and permanent hardness combined. The *permanent* hardness is found by boiling a known volume of the water for some time to remove the temporary hardness and then adding standard soap solution as before. The difference between the total and permanent hardness gives the temporary hardness.

Preparation of Standard Soap Solution.—A solution of Castile soap is made in aqueous alcohol of such a strength that 1 cubic centimetre of it is equivalent to 0·001 gram of calcium carbonate. This is done by weighing exactly 1 gram of pure calcium carbonate (Iceland spar) dissolving it in dilute hydrochloric acid (HCl), evaporating off the acid over a water bath and then dissolving the residue in pure distilled water making it up to 1 litre. One cubic centimetre of this solution contains calcium

chloride ($CaCl_2$) equivalent to 0·001 gram of calcium carbonate; 20 cubic centimetres of this water are then diluted with 50 cubic centimetres of distilled water, and placed in a stoppered bottle for use in standardising the soap solution.

The soap solution is made by dissolving about 10 grams of Castile soap in half a litre of methylated spirit, and then adding distilled water, making the total up to one litre. This soap solution is then run from a burette, 1 cubic centimetre at a time, into the water in the bottle and the bottle shaken vigorously. Soap solution is added in this way until a lather is obtained which lasts for at least two minutes. The soap solution is then diluted with distilled water until 21 cubic centimetres produce a lather with 20 cubic centimetres of the standard calcium chloride solution and the added 50 cubic centimetres of distilled water (the extra 1 cubic centimetre is required to form the lather with the 70 cubic centimetres of distilled water, the remaing 20 cubic centimetres neutralising the calcium chloride in the 20 cubic centimetres of chloride solution).

To estimate the hardness.—Seventy cubic centimetres of the water to be tested are put in a stoppered bottle, and the standard soap solution (prepared as above) added from a burette until a permanent lather is produced on shaking vigorously. Subtract 1 cubic centimetre from the volume of soap solution added (this being the amount required to produce a lather with 70 cubic centimetres of distilled water), and the remainder gives the number of grains of *total* hardness per gallon reckoned as calcium carbonate ($CaCO_3$).

To find the *permanent* hardness counterpoise carefully about 200 grams of the water in a flask, boil for about 30 minutes to remove the temporary hardness, then cool it and make up carefully with distilled water to the original weight. Filter through a *dry* filter paper and titrate 70 cubic centimetres of the filtrate with standard soap solution from a burette as before until a permanent lather is obtained. Subtract 1 cubic centimetre from the volume of soap solution added, and the remainder will give the *permanent* hardness in grains per gallon.

5. Removal of Temporary Hardness.—As already stated, the chief cause of temporary hardness is the presence

of the bicarbonates of calcium and magnesium, which are removed by boiling according to the equations

$$\underset{\text{Calcium bicarbonate}}{Ca(HCO_3)_2} = \underset{\substack{\text{calcium carbonate}\\\text{(precipitate)}}}{CaCO_3} + \underset{\text{water}}{H_2O} + \underset{\text{carbon dioxide.}}{CO_2} \quad . \quad . \quad (1)$$

$$\underset{\text{Magnesium bicarbonate}}{Mg(HCO_3)_2} = \underset{\substack{\text{magnesium carbonate}\\\text{(precipitate)}}}{MgCO_3} + \underset{\text{water}}{H_2O} + \underset{\text{carbon dioxide.}}{CO_2} \quad . \quad (2)$$

It is not convenient in practice to remove temporary hardness in this manner. In Clark's process, slaked lime ($Ca(OH)_2$) is added to the water and precipitates calcium carbonate ($CaCO_3$), converting the magnesium carbonate into magnesium hydrate ($Mg(OH)_2$), which is soluble and will not produce scale in the boiler. These reactions are shown in equations (3) and (4).

$$Ca(HCO_3)_2 + Ca(OH_2) = 2CaCO_3 + 2H_2O \quad . \quad . \quad . \quad . \quad (3)$$

$$Mg(HCO_3)_2 + 2Ca(OH)_2 = Mg(OH)_2 + 2CaCO_3 + 2H_2O \quad (4)$$

6. Removal of Permanent Hardness.—The hardness due to salts other than bicarbonates cannot be removed by boiling at atmospheric pressure, and, as already stated, is called permanent hardness. When permanent hardness is due to salts of calcium and magnesium it can be removed by the addition of washing soda ($Na_2CO_3 + 10H_2O$), which precipitates the calcium and magnesium as insoluble carbonates according to equations (1) and (2).

$$\underset{\text{Calcium sulphate}}{CaSO_4} + Na_2CO_3 = \underset{\text{precipitate}}{CaCO_3} + \underset{\text{soluble}}{Na_2SO_4} \quad . \quad . \quad (1)$$

$$\underset{\text{Magnesium chloride}}{MgCl_2} + Na_2CO_3 = \underset{\text{precipitate}}{MgCO_3} + \underset{\text{soluble}}{2NaCl} \quad . \quad . \quad (2)$$

Washing soda will also remove temporary hardness. Magnesium bicarbonate, for instance, is decomposed in the following manner

$$Mg(HCO_3)_2 + Na_2CO_3 = 2NaHCO_3 + \underset{\text{precipitate}}{MgCO_3} \quad . \quad (3)$$

7. Water-Softening Apparatus.—A large number of different designs of water-softening apparatus are used in practice, in which the chief chemicals used are slaked lime or calcium hydrate ($Ca(OH)_2$), soda ash (Na_2CO_3), and caustic soda ($NaOH$). A mixture of slaked lime and soda ash (or in

some cases caustic soda) in suitable proportions is added to the water to be softened. The lime converts the calcium and magnesium bicarbonates into the insoluble carbonates which are precipitated, the reactions being represented by equations (3) and (4), Art. 5. The soda ash precipitates the calcium sulphate as insoluble carbonate (equation (1), Art. 6), the magnesium chloride as magnesium carbonate (equation (2), Art. 6), and the magnesium sulphate as insoluble carbonate ($MgCO_3$) according to the equation

$$\underset{\text{Magnesium sulphate}}{MgSO_4} + Na_2CO_3 = \underset{\text{precipitate}}{MgCO_3} + \underset{\text{soluble}}{Na_2SO_4}$$

In the Archbutt-Deeley process, used on the Midland Railway, a small quantity of CO_2 from a small coke stove is introduced into the softened water as it is drawn from the softening apparatus in order to prevent any slight deposit of carbonates which might take place in the pipes or injectors when the feed gets heated. The test for any excess of lime which might be in the softened water is to add silver nitrate ($AgNO_3$) solution to a little of the water; a brown precipitate of silver oxide (Ag_2O) shows the slightest excess of lime.

8. Method of Testing Water for Corrosiveness.*— The first thing in testing is to see that the colour of the sediment of the water, as shown in the gauge glass, is neither black nor red. The only colour admissible is slightly dirty grey or straw colour. So long as the sediment of the water is red or black, corrosion is going on, and it must be immediately neutralised by the intelligent use of lime or soda, and frequently scumming or blowing off, the make up being provided by evaporators.

The ordinary salinometer is an instrument for determining the total quantity of solid matter in the boiler water. The apparatus here described gives a convenient and correct method of ascertaining the exact number of grains of chlorine in the water to be tested. It consists of one graduated bottle, one bottle of silver solution containing 4·738 grains of silver nitrate

* This method is reproduced by kind permission from the pamphlet of Messrs. Babcock & Wilcox on Marine Water-Tube Boilers.

to 1000 grains of distilled water, and one bottle of chromate indicator, which is a 10 per cent. solution of pure neutral potassium chromate. It must be clearly understood, however, that this apparatus merely determines the amount of chlorine which the boiler contains per gallon. The solid matter per gallon corresponding to the chlorine is given in the accompanying Table VII.

TABLE VII.—SHOWING PROPORTION OF CHLORINE AND TOTAL SOLID MATTER IN VARIOUS DENSITIES OF BOILER WATER.

Ord. Salinometer Ozs. per Gall. 5 ozs. = $\frac{1}{32}$.	Chlorine Grains per Gall. at 60° Fah.	Total Solids, Grains per Gall. at 180° Fah.	Admiralty Hydrometer Scale.
	1400	2310	
	1350	2227	
5 Density of sea-water	1330	2187	10°
	1300	2145	
	1250	2064	
	1200	1980	
	1150	1897	
	1100	1815	
4			8°
	1050	1734	
	1000	1650	
	950	1567	
	900	1485	
	850	1402	
3			6°
	800	1320	
	750	1237	
	700	1155	
	650	1072	
	600	990	
2			4°
	550	907	
	500	825	
	450	742	
	400	660	
	350	577	
1			2°
	300	495	
	250	414	
	200	330	
	150	247	
	100	165	
	50	82	
	0	0	0°

To make the test.—Fill the graduated bottle to the zero mark with the water to be tested; add one drop of the chromate indicator and shake the bottle; then slowly add the silver solution; keep shaking the bottle. On nearing the full amount of silver solution required, the water will turn red for a moment and then back to yellow again when shaken. The moment it turns red and *remains red,* stop adding the silver. The reading on the graduated bottle at the level of the liquid will then show the amount of chlorine in grains per gallon. For example, if a permanent red colour is shown when the level is midway between 150 and 200, there are 175 grains of chlorine per gallon.

The principle of the process depends upon the fact that if some of this silver solution be dropped into water containing a chloride, a curdy white precipitate of chloride of silver will be formed. If there is also present in the water enough potassium chromate to give a yellow colour, the white precipitate will continue to form as before, owing to the silver having a greater affinity for chlorine than for the chromic acid in the chromate. But, at the moment when all the chlorine in the sample has been converted, the silver will attack the yellow potassium chromate, and chromate of silver will be formed, which is red in colour. The amount of chlorine present is, therefore, shown by the amount of silver solution required to convert it all to silver chloride, and the exact point when the chloride precipitate ceases to form is shown when the chromate indicator turns from yellow to red.

It is not necessary to add the silver solution until the colour becomes very red, as the delicacy of the reaction would be destroyed, but the change from yellow to yellowish-red must be distinct and must not change on shaking. The sample of water to be tested should be neutral, as free acids dissolve the silver chromate. If acid, it should be neutralised by adding sodium carbonate. Slight alkalinity does not interfere with the reaction, but should the sample be very alkaline, it may be neutralised by nitric acid.

Should it happen that the colour does not change within the limits of the graduations, the sample may be tested by

diluting with distilled water. For example, add three parts of distilled water to one part of the sample. If then, on testing the mixture, the colour changes at 200, the number of grains per gallon in the original sample will be four times this reading, or 800 grains.

The chlorine should be kept down to the least possible amount—say, below 100 *grains per gallon*—as the nearer the boiler water is to fresh water, the safer the boilers will be from corrosion.

TABLE VIII.—SHOWING THE AMOUNT IN POUNDS OF LIME OR SODA REQUIRED TO COUNTERACT THE CORROSIVE EFFECT WHICH VARIOUS ADMIXTURES OF SEA WATER WOULD HAVE IN 500 GALLONS (5000 LBS.) OF BOILER WATER.

Lime in lbs. for every 500 gallons boiler water.	Chlorine Test in grains per gallon.	Soda Crystals in lbs. for every 500 gallons boiler water.	Lime in lbs. for every 500 gallons boiler water.	Chlorine Tests in grains per gallon.	Soda Crystals in lbs. for every 500 gallons boiler water.
18·48	1400	104·16	9·24	700	52·08
17·82	1350	100·44	8·58	650	48·36
17·16	1300	96·72	7·92	600	44·64
16·50	1250	93·00	7·26	550	40·92
15·84	**1200**	**89·28**	**6·60**	**500**	**37·20**
15·18	1150	85·56	5·94	450	33·48
14·52	1100	81·84	5·28	400	29·76
13·86	1050	78·12	4·62	350	26·04
13·20	1000	74·40	3·96	300	22·32
12·54	**950**	**70·68**	**3·30**	**250**	**18·60**
11·88	900	66·96	2·64	200	14·88
11·22	850	63·24	1·98	150	11·16
10·56	800	59·52	1·32	100	7·44
9·90	750	55·80	0·66	50	3·72

A boiler containing 500 gallons of water, of which one-fifth is sea water and four-fifths fresh water (and the chlorine test of which indicates about 266 grains per gallon), would, according to the above table, require little more than 3·3 pounds of lime, or 18·6 pounds of soda, in order to make it neutral and therefore non-corrosive (the exact figures would be 3·5 pounds lime, or 19·6 pounds soda). Whereas a boiler containing 500 gallons of water taken direct from the sea would require 17·5 pounds of lime or 98 pounds of soda, in order to bring about the same result.

It is considered, therefore, that while the amounts of lime or soda in the first case are well within practical limits and may be used in the boiler with advantage, those given for pure sea water might cause serious trouble. The amount of scale that would be deposited by the use of 17·5 pounds of lime, would, in itself, be a sufficient objection, and the trouble that might arise from the introduction of 98 pounds of soda into a boiler containing only 500 gallons of water, is well known to every engineer.

It must be clearly understood, therefore, that the table is given merely for the engineer's information. It should not be taken as an instruction to be implicitly followed for the amount of lime or soda to be used with various densities of sea water. This the engineer can only estimate himself by carefully recording the amounts of soda or lime which he is putting in from time to time and the amount of make-up feed that is being introduced into the boiler and by carefully following out the instructions given in Art. 3.

CHAPTER XI

THE PRACTICAL RUNNING OF BOILERS

In order to obtain the best results as regards economy and durability, considerable skill and experience is necessary. The value of the working instructions contained in the Memorandum by the Chief Engineer, Mr. C. E. Stromeyer, M.I.C.E., of the Manchester Steam Users' Association for the year 1909 warrants their inclusion, and they are therefore here in part reproduced in Arts. 1 to 7 by kind permission.

1. " Advice to Boiler Attendants. Stoking, etc.

"*Raising Steam.*—Before lighting the fires see that there is sufficient water in the boiler, also watch the water gauge while raising steam, for the blow-off valve or the feed-check valve may be so leaky as to pass water as soon as there is any pressure. Large temperature differences in a boiler may produce fractures or start leakages. If the boiler cannot be filled with warm water through an economiser, then the firing should proceed very slowly so that the bottom of the boiler may grow as warm as the top. If pressed for time, the boiler may be filled to the top of the water gauges, and when, with rapid firing, the top water has grown warm, discharge the cold bottom water and continue firing.

"*The Coal Bill* for an average Lancashire boiler is about £300 to £600 a year, and more than £100 can very easily be wasted by negligent stoking and by defects which careful inspection by boiler experts can reveal. It is penny wise and pound foolish to ignore so-called trifles which waste fuel or heat during a whole year.

"*Dense Smoke* indicates either that a boiler is overworked, that the draught is insufficient, or that the stoking is badly done;

light smoke indicates efficient stoking. With poor draught the fires should be kept thin and level by firing frequently and on one side of the grate at a time. If necessary, keep the air grid in the fire doors slightly open just after coaling. A continuous admission of much air is wasteful; it spoils the draught and merely dilutes the smoke, but does not remove the nuisance.

"*Mechanical Stoking* may be as efficient as good hand stoking, especially with small coal, and pecuniary savings can be effected if the cheapness of the fuel more than balances the wear and tear of the boiler and of the mechanical stoker. It is not advisable to use mechanical stokers with forced draught if the feed water is either sedimentary or greasy, for then the furnaces are almost sure either to bulge or to groove.

"*Bulged Furnaces.*—The slow bulging which results from grease, scale, or concentrated impure waters, combined with intensely hot furnace temperatures, can only be stopped by removing its cause.

"*Collapses of Furnaces* due to shortness of water are generally more sudden. Misreadings of the water-level due to reflections from the glass shield, and dirt in the gauge glass, frequently lead to these accidents, but negligence on the part of the stoker, stoppage of the feed supply, and leakages past the blow-off cock may also be the cause. As soon as shortness of water is discovered, cool the furnace plates from both sides as quickly as possible. Open the furnace doors to admit much cold air but do not disturb the fires, ease the safety valves and close the stop valves of other connected boilers so as to cause priming in the neglected boiler. The rising froth will help to cool and stiffen the overheated plates. All available feed should be turned on. Exhaustive experiments by the Manchester Steam Users' Association have shown that this procedure does not cause any increase in pressure.

"*Overhauling Boilers.*—Boiler inspection has now been made compulsory, and boiler owners are required to open out, scale and otherwise prepare their boilers in such a way that the Inspector can make a *thorough examination.* Open out and clean the boiler sufficiently often in order that the scale may not grow too thick. Scale is most easily removed before it has come in

contact with air. The boiler had therefore best be cooled with water in it and the scale scraped and brushed off as the water-level is lowered. The flues should be cleaned every three months or oftener. Improved boiler efficiency is obtained if the soot and tarry matter is brushed and scraped off the plates.

"*Manholes.*—Before opening out a boiler the safety valve should be eased and kept open until the manhole and mudhole lids have been taken off, for if there is a pressure or a vacuum in the boiler the lid may be blown off the boiler or into it as the case may be.

"*Fusible Plugs*, if fitted, should be cleaned on the water side and on the fire side every time that a boiler is opened out. The fusible metal should be renewed on the occasion of the annual inspection.

"*Fittings.*—All valves and cocks should be kept in good working condition. They should be thoroughly overhauled on the occasion of the annual inspection, with the exception of *asbestos packed cocks*; these should be overhauled only by the maker or an expert. *Safety valves* and *low-water alarms* should be eased daily to see if they are in working order, and should be overhauled annually. It is a very *serious matter to overload safety valves* or otherwise alter them. If they do not work properly the manager should be informed so that he may send for an Inspector to deal with the matter.

"*Corrosion.*—Corrosion is generally due to air which has been absorbed by the feed water and which cannot be removed. Its action is intensified by common salt and by chlorides of lime and magnesium; its action is diminished by suitable additions of soda and of lime. Purified water, in addition to obviating scale, also reduces corrosion. Excesses of soda or boiler compositions attack brass fittings and cause leaky seams.

"*Idle Boilers* should be thoroughly washed out and dried. Trays with unslaked lime should be placed inside and the boilers should be closed air tight. If the boiler is to stand ready for immediate use, it should be filled with water to which burnt lime has been added, but unless the boiler is one of a battery and is kept warm, it is likely to condense atmospheric moisture on the outside and corrode if filled with water. . . .

"*Steam pipes* should always be drained and kept dry, for it is *extremely* dangerous to admit steam into pipes in which water may be lodging (see Art. 15, Chap. IX.). It is equally dangerous to drain pipes in which there is steam pressure, and yet without pressure it is difficult to drive water out of pipes when it has once got in. Pipes which are so arranged that water can lodge in them either with or without any valves being shut, should be provided with *steam traps* and these should *not be allowed to get choked.* Crackling noises in steam pipes indicate that they contain water and that an *explosion may occur* at any moment. Such cases should be carefully inquired into.

"**Warnings**:—

Don't overload the *safety valves* or tamper with them.

Don't let the *water level* sink out of sight.

Don't allow the *cocks and valves* to set fast.

Don't open the steam *stop valves* hurriedly.

Don't *empty the boiler* while steam is up.

Don't *open manholes* before easing safety valves.

Don't *raise steam* hurriedly.

Don't use unknown *scale solvent* or *compositions.*

Don't *slake ashes* against boiler fronts.

"These 'advices' have been drawn up so as to be easily understood by the boiler attendants, but in view of the probability of points being raised which could not be specially mentioned, the following comments may, it is hoped, prove of occasional use:—

2. "**Raising Steam.**—A very large number of boiler accidents, which are not real explosions, and are, therefore, not reported upon by the Board of Trade, are due to the lighting of fires in empty boilers. The night watchman is not a boiler attendant, and merely does as he has been taught to do with regard to setting away the fires, and if, as happens now and then, the water has, overnight, leaked out of a boiler, the night watchman may either not know the significance of an empty gauge glass, or may perhaps not feel competent to test it and to refill the boiler if empty. He will therefore merely do as he has done before, and put away the fires, and when the fireman

arrives and finds the boiler empty the damage will have been done, and the factory will have to be stopped for a day or two while the leaky seams and rivets are being recaulked.

"Occasionally it happens that a leakage is started during the period of raising steam. For instance, the blow-off cock may not be tight, or the feed valves may be leaky, in which cases water will at first show in the gauge glasses, but as soon as there is any pressure leakage commences, and the furnace tops are then exposed to the heating action of the flames.

"The resulting collapses are generally of a more serious nature than the sprung seams of the previously mentioned cases, because they take place while steam is being generated, and the result is that the boiler in question may have to be put out of use for one or more weeks. These cases are of the nature of unforeseen accidents, with which night watchmen can hardly be expected to deal, and it is desirable to impress them with the necessity of not proceeding with the raising of steam whenever they notice anything unusual about a boiler, for it is far better to have a delay of a few hours, due to there being no steam, than to have to shut down a factory for days because the boilers have become useless.

"The lighting up after a boiler has been out of use for some time should always be done by the regular boiler attendant, because the opening up of a boiler, its overhauling, cleaning and filling, are associated with so many possibilities of which a night watchman is unaware, that he should not be entrusted with the duty of getting up steam for the first time. Pipe and manhole joints may have been broken and not made good again, or not tightly screwed up; safety valves, cocks and valves, may have been overhauled and not properly put together again, etc. It is clearly too much to expect a man who has had no mechanical training to be able to picture to himself what may be wrong if the boiler under his temporary care does not behave as it usually did. Then, too, there is the risk of causing serious damage to a boiler by too rapid firing while the water is still cold, and many leakages and cracks are directly traceable to negligence in this respect. A temperature increase of 180° F. causes steel to elongate one-thousandth part of its

length, and as the upper part of a rapidly heated boiler may be 270° F. hotter than the bottom, it would be one-six-hundred-and-sixtieth of its length longer than the lower part if it were free to move as it liked; but the upper part is linked to the lower part, and both are constrained from adapting themselves to the altering conditions, with the result that stresses are set up which may amount to 10 tons per square inch. If the customary safe working stress of, say, 5 tons, due to steam pressure, is added to the stresses which are produced by unequal expansion, the factor of safety is reduced from, say, five, to less than two, and ruptures are only prevented by the great ductility of the steel of which boilers are now made. In former days, when iron was used, circumferential cracks and seam rips were fairly frequent if steam was raised too rapidly.

"For these reasons several alternative methods of raising steam have been mentioned in 'Advice to Boiler Attendants' (Art. 1). The simplest is, of course, to raise steam very slowly. The conditions of heating are then almost the reverse of those to be found with intense firing, when most of the heat passes through the furnace plates and little through the shell plates in the flues. With light fires this is not the case, for the products of combustion are already relatively cool and retain what little heat they have, even as far as the chimney. They therefore impart almost as much heat to the boiler bottom and sides as to the furnaces, and thus heat the water in the bottom of the boiler, whereas intense fires would only heat the water which is above the level of the firebars and the rest would remain relatively cool.

"If steam has to be raised rapidly in an idle boiler, and if it is provided with an economiser which is being heated by the waste gases from other boilers, it is, of course, very desirable to fill this boiler with the economiser water, for then the time required for raising the water to boiling point is saved and the increase of the pressure can be proceeded with more rapidly. Not only in this case, but in all others, it is advisable to open the safety valves while raising steam in order that the steam may blow off freely as soon as the pressure exceeds that of the atmosphere, for there can be no question but that heat

transmission is enormously increased when circulation of the water is created by the formation of bubbles. This circulation also materially assists in equalising temperatures. A boiler, if it is one of a series having no economiser, might be warmed up, as is done in the case of marine boilers, by blowing steam into the cold water, but the necessary appliances are rarely fitted. In such cases, where steam has to be raised rapidly, a very good plan is to nearly fill the boiler with water and to have bright fires; then the whole body of water above the firebars will grow hot, while that below the furnaces remains cold. As soon as the boiling temperature has been reached, and the safety valves are blowing off steam, the latter should be shut, the blow-off cock opened, and the lower cold water run out till the lower boundary of the upper hot water touches the boiler bottom. The whole boiler will then be of a uniform temperature and the raising of steam can be proceeded with at a fairly rapid rate." . . .

"*Bulged Furnaces.*—Slow bulgings, as might be expected, are chiefly associated with mechanical stokers.* They are also met with in hand-fired boilers, but rarely unless the feed water is sedimentary or greasy, or heavily charged with salts. In the latter case it is generally believed that nothing need be feared until the density of the boiler water is so high as to result in crystallization. This view is erroneous, for there have certainly been cases where salt deposits have formed on the furnace sides and caused bulges and collapses, although the salinity of the boiler water was comparatively low. The danger is naturally greatest in boilers with little circulation, but in the presence of scale, salt deposits are also easily formed, even with good circulation. The most serious cases of this kind occur when, in a boiler having dense water, scale falls off the furnace sides and rests on the boiler bottom, where it gets cemented together by means of the salt. Occasionally new water enters into this caked mass, which gradually grows to several inches in thickness and causes overheating and bulging. In many accidents of this kind the causes are not apparent, for—because the boiler water is not absolutely saturated—any salt cakes which may

* See Art. 16, Chap. III.

have formed are rapidly dissolved away as the boiler cools down. Wherever there is any suspicion of too much salt in the feed water, some water should occasionally be drawn from the boiler (out of a special cock and not out of the gauge glass cocks) and tested for density. This precaution should be taken whenever cakes of salts are formed around leaky boiler fittings, these indicating that there is an excess of salt in the boiler. . . .

"Serious and suddenly *collapsed furnaces* are nearly always due to shortness of water; though there have been cases where scale or grease, combined with intensely hot fires, has been the cause, but as this cannot be known beforehand it is safest at first to treat all cases of sudden collapses as if they were due to shortness of water. If the water level in a boiler sinks below the furnace tops, they become red hot, and grow so weak that they cannot support the boiler pressure, and suddenly bulge or collapse entirely. Obviously, the most natural counter-move is to cool the furnace plate and thereby strengthen it. This can very readily be done by opening the furnace doors to admit cold air—the fires being left undisturbed—and by easing the safety valves and closing the stop valves on adjoining boilers. As the single injured boiler, instead of several, is now supplying steam to the engines, violent ebullition will be set up, and if the boiler is not already quite empty, froth will rise and cover the furnace crowns and cool them. Above all, the rapid discharge of steam through the safety valves and to the engines will quickly reduce the boiler pressure and either prevent any further mischief being done, or, if a rent should occur, the contents of only one boiler instead of several will be available for doing harm. The feed should also be turned on full, but this means of raising the water level is too slow to be of much use in covering the plates if bulging has actually occurred, but it helps, especially if the delivery pipe is above the water level, to reduce the steam pressure, and it minimises the risk of the furnaces tearing.

"In cases of shortness of water without furnace bulging, there should never be a moment's hesitation about rapidly cooling the furnace plates. Formerly, it was held that by bringing cold water or froth into contact with a red-hot plate

explosive pressures would be generated, but the costly experiments carried out by the Manchester Steam Users' Association on a large boiler have conclusively demonstrated that even if quite cold feed water is sprayed on a red-hot furnace top no increase of pressure, but only a decrease takes place. The older view was probably invented in order to account for explosions the real causes of which had been carefully obscured. The steel used for modern furnaces is so mild that it readily stands the test of being heated to redness, quenched in water and then bent double. There need, therefore, be no fear of cracked plates.

"Most large boilers are provided with low-water and high-pressure safety valves, which blow off steam, relieve pressure, and give the alarm as soon as the water level sinks too low. They close again as soon as normal conditions are attained. Fusible plugs are less satisfactory, and should only be applied in small boilers, for, when acting correctly by blowing out in cases of shortness of water before bulging takes place, they put the boiler entirely out of use until a new plug has been fitted, and very often, if they are not properly scaled, they blow out without the water being low. Then also, if the alloy is not renewed every year its constituents separate from each other and the melting point is raised too high to be of use in cases of danger."

4. "**Overhauling of Boilers.**—The nature and frequency of general overhauling and cleaning depends largely on the sedimentary nature of the feed water. If purified water is used, a boiler can generally be run for three months of ordinary working without fear of corrosion or serious scale occurring. Boilers which are fed with naturally pure waters are even less liable to scale, but may in extreme cases in some districts have to be examined every three months for fear of corrosive activity. Boilers which are fed with sedimentary waters do not need this frequent inspection, but they may have to be cleaned out, say once a month, and in such cases water purification should be resorted to.

"The scaling of boilers is often very expensive, more particularly if the scale is hard and adhesive, and where this is

the case the plan should at least be tried, provided the necessary time is available, of cooling the boiler thoroughly without emptying it; the boiler should be entered when the water is just about level with the furnace tops, and the scale brushed or scraped away as the water level is slowly allowed to sink. This practice is based on the experience that boiler scale does not harden until it comes in contact with air.

"*Soot and tarry matter* should be brushed and scraped off the boiler plates as often as possible, for these substances are amongst the worst conductors of heat, and unlike scale, they settle on those parts of a boiler where the products of combustion are coolest. It is for this reason that economiser tubes have to be scraped continuously, for if this scraping is stopped, soot and tarry matter quickly coat the surfaces, and the economiser efficiency is very rapidly reduced. If the feed water is sedimentary, most of the scale forms on the furnace plates, because here evaporation is most rapid and a maximum of scale is produced. When this scale has grown so thick that it offers considerable resistance to the passage of heat from the flame to the water, less heat will be transmitted and the flame will remain hot longer than when the plates were clean and cold. The flames or hot gases will therefore now extend further beyond the firebridge than they did at first, and the plates beyond the bridge will also receive their coating of thick scale, but even now the waste gases leaving the boiler side flues will be at about the same temperature as before, because little scale has been deposited where the waste gases were coolest. This means that scale in a boiler affects the distribution of evaporation, but it has little influence on the total amount. It is different with regard to soot and tarry matter; they do not get deposited on the hot furnace plates, which therefore remain clean and efficient for heat transmission, but are plentifully deposited on the cooler plates where the difference of temperature between the gases and the water is low. But it is just here that it is highly desirable that the heat-transmitting properties of the plates should be of the best. Thus if, as is probable, one-third of the total heat is transmitted through the shell plates of a clean boiler, then this third may be reduced, say,

to one-sixth when these plates are covered with tar and soot, and this represents a loss of 17 per cent. If at the same time thick scale is being formed on the furnaces, and a higher duty, say 50 per cent., is thrown on the shell plates, then the soot coating is likely to reduce transmission to, say, one-half, and there will be a loss of 25 per cent. It will thus be seen that sedimentary water alone and smoky coal alone are not so bad as when they are found in company."

When preparing a boiler for cleaning or inspection it should be cooled and emptied as gradually as possible. The best method to adopt is to close the stop valve, draw the fires, and allow the boiler to stand several days if needs be until both it and the brickwork setting are cooled. The top tap of the water-gauge fitting should be opened, or the safety valve lifted off its seat, when the pressure has fallen to nearly atmospheric in order to prevent a vacuum being formed in the boiler, otherwise a weak boiler may collapse if subjected to external atmospheric pressure. When cool, the water is run off through the blow-off valve, and the boiler and flues opened. By cooling slowly in this manner straining of the seams is prevented.

If lack of time prevents the above method from being followed, the injurious straining of the seams produced by cooling rapidly may be reduced by pumping cold water into the boiler as fast as the hot water is run out. By this method the boiler shell below the water level is cooled uniformly together with the water, and when cold, the boiler is emptied through the blow off as before. The boiler should never under any circumstances be blown down and emptied under pressure and then cold water pumped in to cool it. The sudden strains which would be set up by the cold water running on the hot plates may fracture the plates through the lines of rivet holes forming the seams; such fractures are difficult to detect and may lead to explosion when the boiler is again worked.

The above remarks apply more particularly to boilers of rigid construction, *i.e.* boilers of the smoke-tube type. Water-tube boilers on account of their freedom for expansion and contraction may be blown down and cooled rapidly without injury.

5. "**Manholes.**—There have been many accidents due to the opening of manholes before the safety valves have been eased, and it is thought advisable to give a word of warning. Supposing that there is a slight pressure of, say, 2 pounds per square inch in a boiler, while the manhole bolts are being unscrewed, then it is almost certain that the lid will be blown up after the bolts have been taken out and as soon as the jointing material gives way. The lid may either fly into the air and fall perhaps on the attendant, or if it is held down by a few loose bolts, steam and hot water will be squirted out horizontally and scald him. Another possibility is that a vacuum has been formed in a boiler after cooling down, then, on unscrewing the lower manhole dogs, it may happen that the door will be suddenly sucked in, and the attendant blown against the opening. It must be remembered that the power for mischief which is represented by a vacuum of the amount possible in a Lancashire boiler is large enough to do a great deal of mischief." . . .

6. **Fittings.**—Safety valves and other safety appliances, including water gauges, should be tested daily, and oftener, for so much depends on their correct working that absolute certainty about their condition should be aimed at. Any tampering with safety appliances is, of course, a most serious matter and should not be permitted—it may lead to an accident. In view, however, of the impossibility of refitting asbestos packed cocks without special appliances, these cocks need not be opened out on the occasion of an annual overhaul, but must be tested in order to see that they work freely.

7. "**Idle Boilers.**—Cases of severe corrosion are often reported where nothing injurious can be detected in the feed water, but on close examination it very generally transpires that these boilers are occasionally laid off without being properly prepared, and they are then exposed to several serious attacks. There is, firstly, the probability of a severe internal attack due to moisture which may have remained in the boiler and to the abundance of carbonic acid to be found in stokeholds. Added to this, most waters contain at least traces of common salt and

of chloride of magnesia both of which are very powerful corrosive agents in the presence of moist air. Possibly, too, the sulphurous acid fumes which are disengaged whenever red-hot ashes are slaked, find their way into the adjoining empty boiler, which is then subjected to the combined attack of atmospheric oxygen, carbonic acid, moisture, salt, chloride of magnesia and condensed sulphurous acid vapours.

"Quite recently some severe pitting of boiler tubes had to be investigated, the assertion being made that sulphuric acid from the waste gases had somehow got into the boiler water, for the rust taken from the pit holes was strongly charged with sulphate of iron. There could certainly be no doubt about this fact, but on visiting the works it was found that whenever a tube had pitted through, it was drawn out of its boiler and laid aside in a space into which there was a strong leakage of waste flue gases, which, as usual, were heavily charged with sulphurous acid. This acid had been absorbed by the spongy rust of the pit holes, where it was converted into sulphuric acid. The scrapings off any tube which had just been drawn contained no traces of sulphuric acid. This simple case shows the nature of the activity of the noxious vapours to be met with in stoke-holds; they are absorbed by moist rust which lodges on the plates.

"To prevent corrosion in idle boilers these should be emptied and dried, trays of unslaked lime should be placed inside, and then all the manholes and other openings should be hermetically sealed. The lime will very soon absorb any moisture, carbonic or sulphurous acid which may be in the boiler, and any remaining salt or chloride of magnesia, being dry, can do no harm.

"Formerly it was a very common practice to fill idle boilers with water which had been charged with soda or lime, and no doubt internal corrosion can by these means be prevented, but the chances are that very severe external corrosion will now occur. The water in the idle boiler being cold, its flue surfaces, where they are exposed to leakages from the flues of adjoining boilers, will condense moisture highly charged both with carbonic acid and sulphurous acid, and external corrosion will

progress rapidly. If, for instance, the atmosphere is humid, due either to fog or rain, then the flue gases of adjoining boilers will contain not only this moisture but a large addition due to the combustion of the fuel, and when these gases come into contact with a cold surface and are cooled down again to the temperature of the outside atmosphere, all the added moisture, which may amount to 3 per cent., will be condensed as mentioned above, and cause corrosion. This is not a mere speculation but an experience which is frequently confirmed with economisers. With them the inlet ends of the pipes, which of course contain only cold water, act as condensers of the flue gas moisture and waste away very rapidly. The remedy in these cases is to add a little hot water to the cold feed so that the temperature of the inlet pipes is above that of the dew point of the heavily laden flue gases.

"In some works it is necessary to have one or more boilers always ready for starting, and then of course they had best be kept filled with water containing a little caustic soda, but in all such cases it would seem desirable to devise some flue bye-pass arrangement whereby the idle boiler could be sufficiently warm to prevent condensation."

8. Instructions for the Preservation and Working of Water-Tube Boilers.—The following instructions are issued by Messrs. Yarrow & Co., Ltd., for the working of their marine boiler, but together with the advice already given above are in the main applicable to other water-tube types.

"*Precautions when lying up.*—For the better preservation of a water-tube boiler when lying up, it is advisable that it should be emptied and drained of water and thoroughly washed out internally with clean fresh water. Ashes and any accumulation of soot should be removed from the tubes and tubeplates. This is of the utmost importance, because if moisture becomes absorbed by the dirt which collects on the heating surfaces, corrosion will soon commence, and, when once started, will increase rapidly. The outside of the tubes may then be cleaned by means of a hose with as good a force of fresh water as is available. The casing should be carefully swept on the inside.

"A small coke fire should be lit in a suitable portable receptacle, which may be placed in the ashpan to thoroughly dry both the inside and outside of the boiler, and for this object a portion of the firebars must be removed so that the coke fire can be kept far enough from the tubes to avoid overheating them. The manhole and mudhole doors being off, the vapour formed will escape.

"*Internal Examination.*—A governing feature of the Yarrow boiler is the facility for examination of every portion internally, and this can be effected from the steam chest or drum, access to which is gained by simply removing the manhole cover. When the boiler is dry, a brush or other appliance should be passed through each tube. Every tube, being straight in the Yarrow boiler, can be examined and cleaned throughout its entire length. The examination can best be done by means of a small electric light attached to a flexible conductor and fed by storage cells or a dynamo and lowered down the tube. By this means, any obstruction or serious corrosion becomes visible, and should scale or obstruction be found to exist it should be at once removed. The tubes being straight this offers no serious difficulty.

"*Preservation when lying up.*—If a boiler is intended to lie up for a lengthened period, some quicklime in suitable trays should be put into the lower and upper drums. The drums should then be closed up to exclude the air, care being taken to remove the lime before again filling the boiler with water. The object of the quicklime is to absorb any moisture that might remain in the interior of the boiler.

"Another reliable practice when laying a boiler up is, after it has been thoroughly washed out, to close up all manhole and mudhole doors, and to quite fill the boiler with clean water, adding 9 pounds of common washing soda to each ton of water, this soda being dissolved in the water before it is put into the boiler. Care should be taken before again starting the boiler under steam to thoroughly empty it. A coke fire, such as before described, should be provided from time to time, so as to keep the boiler at a temperature slightly in excess of the surrounding atmosphere, otherwise moisture may collect on the outside surfaces and cause corrosion.

"If the boiler is to lie up for a lengthened period it is also very desirable that the brickwork should be removed and only replaced when required.

"*Raising Steam.*—When it is intended to raise steam the boiler should be filled with water to the top of the gauge glass, and 1 or 2 pounds of ordinary lime per 1000 gallons should be added in the form of milk of lime. Care must be taken that the lime is well mixed before being put in the boiler, and the lime water should be passed through a fine strainer.

"*Joints in Water Pockets.*—A bolted joint is provided to the lower water pockets of the smaller Yarrow boilers (p. 219). The joint is made with asbestos metallic sheeting $\frac{1}{10}$ of an inch thick. Before breaking this joint the weight of the boiler should be carried on lugs provided for that purpose at each end of the lower tube plates. When re-making the joints of these water pockets, after having screwed the joints up as tightly as possible, steam should be raised to 10 pounds per square inch to thoroughly warm the boiler, and the bolts in the joint finally tightened up. It is only contemplated to break these joints in case of important repairs.

"*Precautions when in Use.*—When working, every opportunity should be taken to shut down each boiler in rotation in order to examine the brickwork and clean the tubes inside and out. Free use should be made of the electric lamp by passing it down the tubes to ascertain that they are clean and perfectly free from obstruction, and that there is no incrustation. The two or three rows of tubes nearest the fire require more careful attention than those which are further from it, and if any accumulation of sediment is found it should be removed before the boiler is started again.

"*Feed Water.*—In regulating the feed, the check valve should be altered very little at one time. Careful adjustment is required at first, and when once set to a suitable area of opening little further attention is required. No oil should be allowed to get into the boiler. If any oil is used for the internal lubrication of the machinery, it should be mineral oil (Art. 3, Chap. VII.), and in many engines it is found that oil can be dispensed with altogether, which is very desirable. As

little oil as possible should be used for lubricating the piston rods, because a certain amount of oil invariably finds its way in to the interior surfaces by this means. Special care should be taken that the auxiliary engines are not of such a character as to involve the use of oil for internal lubrication. The auxiliary engines should be run without any internal lubrication whatever, and if any lubricators are fitted they should be removed.

"The water used in the boiler should always be distilled, and only when unavoidable should it be obtained from the shore, as that will often lead to the formation of scale. Tests should be made from time to time to ascertain that there is no acid contained in the water in the boiler, and not only should it be alkaline, but it must be definitely so. For this purpose from 1 to 2 pounds of ordinary lime per 1000 I.H.P. should be pumped daily into the feed as milk of lime, or more if found necessary to ensure the water being decidedly alkaline. No sea water on any account should be allowed to get into the boiler, and for this reason care must be taken that the condensers are tight, that the evaporator does not prime, and that all sea connections are properly shut. If, however, sea water does get into the boiler, double the ordinary quantity of lime should be used with the feed, the fires must not be forced, and the density kept as low as practicable.

"*Ashpit and Fire Doors.*—The ashpit doors must always be kept shut and properly secured so that in the event of a boiler tube bursting or steam escaping suddenly through any other cause, it may not find its way into the stokehold. For the same reason the fire doors should be kept closed, except when stoking. In the event of a serious leakage of steam the fan should be immediately turned on to force the escaping steam up the funnel; the stokehold should be closed, the pumps turned on full speed and the fire extinguisher put into operation. The fire extinguishing apparatus should be tested from time to time to ascertain that it is in perfect working order.

"*Casing Joints.*—The casing joints must all be perfectly tight because leakage of air will cause rapid destruction, and care

must be taken not to allow any large accumulation of ashes or coal-dust in any portion of the casing.

"*Tube Plugs.*—In the event of a tube giving way the ends should be closed by plugs provided for the purpose. No appreciable reduction of efficiency would be found even if after lengthened service ten per cent. of the tubes are inoperative.

"*Stoking.*—The fires should be adjusted so that no appreciable quantity of smoke issues from the funnels, because if there is much smoke soot will be deposited on the outside of the tubes and will clog the spaces, thus reducing the efficiency of the boilers. On the average, a thickness of fire of from 5 to 6 inches has been found with Welsh coal to be suitable. When charging the furnace the coal must be thrown on in the exact places where required, and not piled up at the front end of the grate and afterwards pushed back, as is customary with ordinary marine boilers. (See p. 59.)

"*Explosion.*—When opening the boilers or any part connected to them, such as pipes, condensers, etc., great care must be taken to prevent any open light being near, as the explosive gases formed in the boilers may catch fire and cause serious injury."

9. Automatic CO_2 Recorder.—Although by good stoking and by the use of a suitable fuel the quantity of air supplied per pound of fuel can be kept down to a suitable value, it is also important to remember that there should not be any leakage of air through the brickwork into the flues. Air leaking in to the flues (when forced draught is not employed) between the boiler and the chimney is frequently a cause of inefficiency, and no matter how skilful the stoker may be, the loss due to this leakage will always be present if the brickwork is not kept in perfect condition; by reducing the temperature of the gases in the chimney it reduces the draught, and therefore not only indirectly reduces the temperature of the fire but also prevents the maximum output being obtained from the boiler. The presence of this leakage makes itself felt by reducing the percentage of CO_2 in the flue gases, and it is only by having a continuous record of the amount of CO_2 present that a continuous control can be kept over the flue and boiler settings which are so apt to give trouble unexpectedly and to crack in the most inaccessible and darkest

places. Even when the most careful precautions are taken to prevent this air leakage (Art. 11, p. 253) the use of a reliable automatic CO_2 recorder cannot fail to be beneficial, for it acts as a check on the stoker and gives him something definite to work to. In addition to being a control on the stoker it also acts as a check on the quality of the fuel supplied by the coal merchant.

There are several reliable CO_2 recorders on the market, but lack of space forbids the description of more than one type.

The Bi-Meter CO_2 Recorder.—The general appearance of this apparatus as made by the Cambridge Scientific Instrument Co., Ltd., is shown in Fig. 131, while Fig. 132 is a diagrammatic sketch of the internal connections. The apparatus consists of two gas meters M_1 and M_2, an absorption box E, a water-suction pump B and a recording mechanism FG.

The water-jet suction pump B, using about $2\frac{1}{4}$ gallons of water per hour, draws about $1\frac{1}{2}$ cubic feet of the flue gas through the instrument per hour. The gas, entering at A is cooled in the first chamber of the cooler K and is then measured in the meter M_1. The CO_2 is then extracted from the gas in the absorption chamber E containing lime; and, since during this chemical process the remainder of the gas becomes heated, it is again cooled to its former temperature by being passed through a second chamber in the cooler K. From the cooler the gas is led to the second meter M_2 to be again measured, and is thence allowed to escape to atmosphere by way of the aspirator B and the water vessel C.

The water which is employed for the working of the instrument enters through the cock D, and flows through the cooler K, into the aspirator B. It there draws in the flue gas and the mixture of water and gas (from which the CO_2 has been removed) passes into the water vessel C from which the water escapes through an overflow drain pipe H, and the gas bubbles into the atmosphere.

The two gas meters are filled with oil, and are so arranged that, when no absorption takes place, the meter M_2 runs about 4 per cent. slower than the meter M_1. Thus when no CO_2 is being absorbed the pen is made to record lines about 3 or 4 mm.

in height, and adjustment must be made so that the upper ends of these lines should lie on the zero line of the chart. When this is secured, the apparatus, on connecting up the absorption chamber

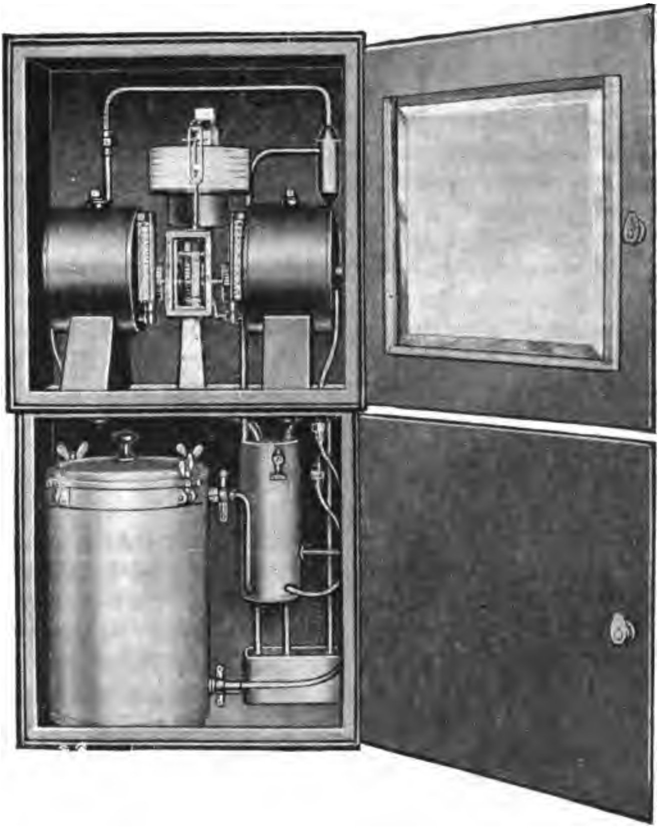


FIG. 131.—General view of the bi-meter CO_2 recorder.

in position, records the percentage by volume of CO_2 contained in the gases which are under test.

The recording pen is actuated by means of a differential drive F, operated by the meters M_1 and M_2. On an average from 20 to 25 analyses may be recorded per hour, the number being

dependent upon the volume of the flue gases passing through the instrument. The number may be regulated by adjusting the tap or valve P placed near the aspirator B, and should be so adjusted that, with a clear recording pen, the lines do not run into one another. When the percentage of CO_2 is high, it is preferable to keep the number of analyses per hour fairly low in

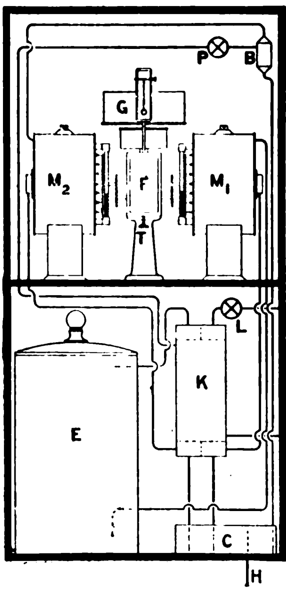


FIG. 132.—Internal connections of the bi-meter CO_2 recorder.

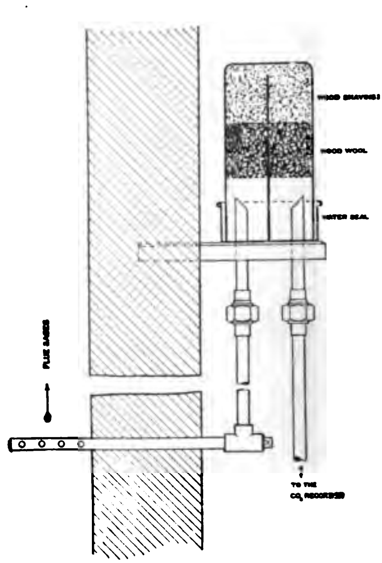


FIG. 133.—Diagram of soot filter connections.

order to avoid exhausting the lime in the absorption chamber E too quickly.

In order to remove any solid matter from the gases, a soot filter is interposed between the flue and the recorder. The upper portion of the soot filter is filled with wood shavings and wood wool as shown in Fig. 133. It is then placed in the lower part, which is filled with water so as to form a water-seal. In order that this water may not evaporate it is covered with a layer of oil poured on to its surface. It is important that this soot filter

should be connected immediately above the intake pipe for convenience in cleaning out the piping which connects it to the flue.

A portion of the actual record obtained from this automatic recorder is shown full size in Fig. 134.

The maximum theoretical amount of CO_2 by volume being about 21 per cent. of the flue gases, a figure of about 12 per cent. may be regarded as good practice.

10. The Purchase of Fuel.—In Germany and the United States the calorific value of a coal usually fixes its price, and a

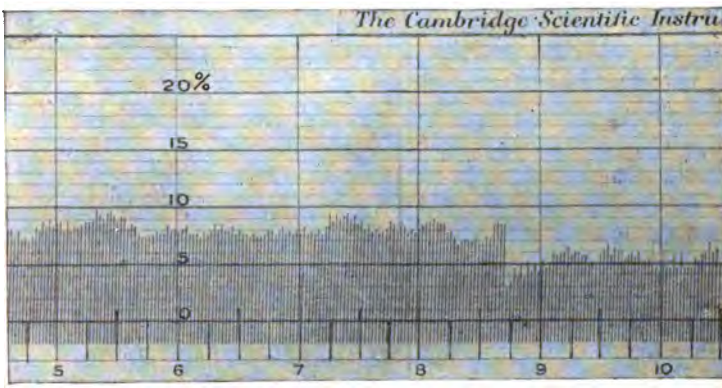


FIG. 134.—A portion of an actual record.

sum of money is laid out to purchase a certain number of given B.Th.U. rather than a certain *weight* of coal. On the other hand, this method of buying coal is not usually adopted in Great Britain. Since the function of a steam plant is to convert as much as possible of the heat in a fuel into a form in which it can be more usefully employed, it would appear at first sight that the most economical results would be obtained when the fuel is bought on a calorific basis. Mr. C. E. C. Shawfield * has stated that by adopting this method and installing the requisite testing apparatus in his station at Wolverhampton, he has been

* In the discussions on *The Purchase of Fuel* at a meeting of the Incorporated Municipal Electrical Association (1911).

able to effect a saving on the coal bill alone of £500 per annum on a total fuel consumption of from 20 to 50 thousand tons a year, owing solely to the improvement of the quality of the fuel.

There, are, however, several objections which may be raised against the purchase of fuel on a purely calorific basis, and it will be well to consider the principal factors which require consideration when deciding on the best coal to use in any particular case.

The amount of water evaporated per pound of fuel depends on the calorific value of that fuel and the boiler efficiency for that particular fuel. It must be remembered, however, that the efficiency of the boiler depends not merely on the boiler itself, but also upon the stoking and class of fuel independently of its calorific value. The percentage of ash, moisture, and sulphur are also important items, and in addition, there is the cost of carriage to be taken into consideration. As regards the boiler itself, the best fuel to use will be the one which gives the highest efficiency under everyday working conditions. This does not necessarily mean that the best fuel to use in a given case is the one of highest calorific value; the locality of the power station is an important factor to be considered because of the cost of carriage.

A cheap fuel of low calorific value, say, 10,000 B.Th.U. per pound, might be best in some cases, whereas in the case of a station a long way from the coalfields it might not pay to buy coal of less calorific value than, say, 14,000 B.Th.U. per pound. Another important item to be considered is the method of firing adopted. If self-cleaning mechanical stokers are used, an inferior class of fuel can be used with much less heat loss than would result from the frequent cleaning of the fires necessary with hand-firing.* In some cases a mixture of several different qualities of coal might be found to give the most economical results.

A serious objection to the purchase of fuel on a calorific basis lies in the difficulty experienced in determining its average calorific value. A very small quantity (about 1 gram) is used in a

* Mr. C. E. C. Shawfield, in the discussions on *The Purchase of Fuel*, mentioned in the preceding footnote.

laboratory test and it is a difficult matter to ensure that this small amount is a representative sample of a large consignment of coal (see Art. 14, Chap. II.).

Again, two different kinds of coal may have practically the same calorific value, but when burned under the same boiler may give quite different results. From this alone it would appear that the laboratory test in itself is not conclusive, and that boiler tests (Chap. XII.) are more reliable.

If the coal owner, or agent, guarantees a certain calorific value when tendering for a contract, he will increase his price. This *might* counterbalance the saving in works' costs, with the result that the net gain to the engineer might be negligibly small. Also the coal owner would doubtless prefer to guarantee his calorific value at the pit because of the deterioration which coal is liable to undergo when exposed to the atmosphere for an appreciable time during storage or carriage to a distant works.

It would appear that even if the purely calorific basis be rejected, it is extremely desirable that systematic tests should be made of all fuel that is used. The coal supplied under contract would undoubtedly be much better and more uniform in a quality if a constant watch were kept upon the ash, the moisture and the calorific value. Even laboratory tests pure and simple might form adequate protection for the small consumer, but it would be safer to carry out in addition, frequent boiler trials. Large consumers would greatly benefit by having their own laboratory in which the coal would be tested for the percentage of ash, moisture, sulphur and volatile matter, in addition to the usual test for calorific value in the Bomb Calorimeter, and it seems only reasonable to suppose that in the long run they will inevitably tend more and more to buy coal on a thermal basis, although, as mentioned in the above short discussion, there are many objections to such a course and the specified conditions should admit of some latitude.*

* See also a paper on the Systematic Purchase of Coal, by Messrs. F. R. Hutton and J. L. Pultz, in *Power*, vol. 35, p. 285, 1912.

CHAPTER XII

STEAM BOILER TRIALS

1. The Object of a Boiler Trial.—The majority of boiler trials are carried out for a commercial purpose, *i.e.* to prove whether the performance of a boiler is up to the guarantee which has been given by the maker to the buyer, and in such cases it is only necessary to weigh the fuel burned and to determine its calorific value and also to weigh the water evaporated. These measurements are all that is wanted to enable the thermal efficiency of the boiler to be estimated. In order to carry out a complete investigation of the performance of a boiler however, it is necessary to make, in addition to the above, measurements of the losses which occur, because these will serve as a check upon the accuracy of the trial. For instance, if the results of a boiler trial are summed up in the statement that its efficiency is, say, 80 per cent., such a result may or may not be correct; if however, a heat balance is drawn up giving the measured losses in addition, the result is much more definite and it can be examined in detail, so that such an efficiency, although high, might be accepted. There is, of course, always one item in the heat balance which is obtained by difference, and includes the unmeasured loss due to radiation and all the errors of the observations made. If in conjunction with a high efficiency, say 80 per cent., the "unaccounted for" item is too low, say below 5 per cent., then the accuracy of the trial would be open to severe criticism. If the "unaccounted for" item is too high, say 20 per cent., it would again indicate that greater accuracy is desirable.

The whole question of boiler trials was investigated by a Committee of the Institution of Civil Engineers, and in 1902

were finally drawn up standard forms to include tests made for scientific and also for commercial purposes.* By permission of the Council of the Institution of Civil Engineers these are reproduced on pp. 359–373, together with the chief instructions :—

2. Measurements required for a Complete Boiler Trial.—The various measurements required for a boiler trial are divided by the Committee's Report into two groups as follows :—

To determine the Thermal Efficiency of a Boiler.

(1) Rate of fuel consumption.
(2) Sampling of the fuel and determination of its calorific value and of the moisture in it.
(3) Rate of water evaporation.
(4) Measurement of steam pressure.
(5) Determination of the dryness of the steam.
(5*a*) If there is a superheater, the temperature of the steam.
(6) Measurement of the feed temperature.

To determine the Losses.

(7) Measurement of the temperature of the flue gases.
(8) Collection and analysis of the flue gases.
(9) Sampling and weighing of ashes.
(10) Chemical analysis of ashes, or determination of their calorific value.
(11) Measurement of air pressure.
(12) Radiation and leakage.

In order to obtain good results from a boiler on trial special attention must be paid to stoking, if the boiler is hand fired, because however good the boiler may be, it is evident that with unskilful firing a low efficiency will result (Art. 6, Chap. III.). The method of starting and stopping the trial and the duration of the trial are also of great importance. Before describing the methods adopted in making the above measurements it will be as well to consider these two items.

* *Proc. Inst. C.E.*, vol. cl., p. 218. The report may also be obtained separately from Messrs. W. Clowes & Sons, Ltd., Duke Street, Stamford Street, S.E., and must be referred to for the full instructions.

3. Starting and Stopping the Trial.—With reference to this the Committee say: "The best way of starting and finishing a boiler trial must be decided by the principal observer on the spot, because it is governed by the conditions under which the boiler has to work, but one of the following three methods may generally be used.

"A. *First Method*, applicable when the rate of evaporation can be kept uniform throughout the trial. The fires having been cleaned and made up a short time before it is intended to start the trial, and everything being in the ordinary working condition, the principal observer gives the signal to commence firing with fuel from the first bag or box of coal; at the same instant the feed pump should be started upon the first measured tank of water, and the level of the water in the boiler and in the feed water tank should be noted.

"The time at which the above three things are done is taken as the *starting-time of the trial*. Throughout the trial, the time of commencing and finishing each bag or box of coal and each tank of water must be carefully noted, and also the level of water in the gauge glass. While the trial is proceeding it is advisable if possible to plot both the fuel and feed measurements to a time base, as in this way a most valuable check on the observations is obtained. . . .

"The fuel measurements may be stopped after about n hours' run at the moment any bag or box of coal is finished, n being a multiple of the time it is possible to run the boiler without cleaning the fires. The grates should be cleaned and the fires made up previous to the stoppage as they were previous to the start, and similarly at about the same time the feed measurements may be stopped when any tank is finished. It is desirable that the water-gauge level should be the same at the end as it was at the beginning; if this is not possible then a correction for the difference of level must be made. The weight of water corresponding to a difference of level of each $\frac{1}{10}$ inch in the boiler should be determined before the trial begins in order that the correction to the feed measurements may be plotted as the trial proceeds. . . ."*

* "It should be remembered that *when the final is higher than the*

"B. *Second Method.*—The fires are lighted long enough before the intended time for commencing the trial to bring everything into uniform conditions. If the boiler has been working for some time, all the fires are thoroughly cleaned and got into good order, and the rate of evaporation and the corresponding feed are then adjusted as nearly as possible to the intended rate. About a quarter of an hour before the time fixed for the start sufficient water is put into the feed tank to last for this interval of time; the firing of the boiler is stopped, any coal lying near the boiler on the firing plate being swept up and removed. The fire is now examined carefully from time to time and raked up to close any holes, and the pressure gauge is carefully watched and noted at short intervals, until, at a fairly distinct moment, the pressure begins to drop rapidly. This is the moment at which the heat supplied from the furnace becomes insufficient to maintain the rate of evaporation, and this instant is noted as the beginning of the trial, and the following operations are then effected:—

(*a*) The feed-pumps are stopped.*

(*b*) The height of water in the boiler gauge-glass is noted. (It is a good plan to make a mark on a smooth piece of wood fixed to the back of the gauge-glass).

initial level the weight of water to be deducted from the measured feed is not the weight of the layer of water contained between the two levels, but an equivalent weight which, if converted from water at feed temperature into steam at boiler pressure, would have absorbed the same quantity of heat as was required to heat the layer of water between the gauge levels from the temperature of the feed to the temperature of the steam, but not to evaporate it. If W be the weight of water between the two gauge levels t and T the temperatures of the feed and steam respectively, L the latent heat of evaporation, commonly called the latent heat of 1 pound of steam at temperature T, and w the equivalent weight to be deducted from the feed, then

$$w = \frac{W(T - t)}{T - t + L}$$

When the final level is lower than the initial the weight to be added to the measured feed is of course the actual weight W of the layer of water contained between the two gauge levels at the temperature T."

* If there is an economiser the stoppage should be of very short duration to prevent the feed from getting too hot.

(*c*) All ashes are cleaned out.

(*d*) The first bag or box of coal is emptied on to the floor and stoking is recommenced.

(*e*) The level of the water in the feed tank is noted, and the feed pumps are again started.

"Throughout the trial every endeavour should be made to keep the rate both of evaporation and of the feed constant, and to maintain a constant pressure in the boiler with the least possible variation of level in the height of water in the gauge-glass. To attain this, great care is needed throughout the whole of the experiment. Near the end of the trial, the last bag of coal having been emptied on to the floor and finally fired (special care must be exercised to maintain the same rate of evaporation), and the water height in the gauge-glass having been brought back to what it was at the beginning of the trial by regulation of the feed, the pressure gauge is again carefully watched, and the instant at which it begins to show a notable fall is taken as the time of the finish of the trial. It may be assumed that at this instant the furnace again ceases to supply sufficient heat to maintain the working pressure at the rate of evaporation which has been used during the trial, and that therefore the condition of the fire and of the boiler generally is the same as it was at the beginning of the trial.

"It will always be found best when running a trial by this method not to inform the stokers that it is intended to finish the trial at any given time or they will probably unconsciously slacken the stoking. It will be noted that by this method the precise moment of ending the trial is not predetermined, but it should preferably be made *later* than a previously fixed hour." . . .

"C. *Third Method*, applicable when the boiler can be run at approximately the normal rate for some time previous to the commencement of the trial:—

"The boiler having been at work at full pressure long enough to heat the brickwork (if any) to the normal working temperature, the principal observer should order the fires to be cleaned, and note the time of giving the order. This order should be given as short a time before the commencement of the trial as will allow of the grate being cleaned, and the

fires made up and burnt through before the starting time; half an hour to an hour will generally suffice, but of course the time depends on the number of furnaces, the adhesiveness of the clinker, and the number of stokers available. . . .

"As soon as the fires have burnt clear and as low as is compatible with keeping up the steam pressure, the principal observer should again note the time, and blow a whistle or ring a bell to notify the commencement of the trial to his assistants, who will immediately note the water levels in the gauges and the steam pressure. The principal observer himself should then take a rake or a specially made tool, level and gauge each fire in several places and note the average thickness on each grate. If the fires be not more than 3 inches or 4 inches thick, the duration of the trial necessary to attain any required degree of accuracy may be halved by pushing the whole of the fire on to the back half of the grate before gauging the thickness. The weight of fuel in pounds on each grate may be taken as $A \times T \times 2{\cdot}5$ for slack, or as $A \times T \times 1{\cdot}7$ for large coal, where A is the area in square feet of the part of the grate covered by the fuel, T the average thickness of the fuel in inches.

"About half an hour or an hour before the intended conclusion of the trial, according to the time occupied in preparation before the commencement, the principal observer should note the time, and then superintend (*a*) the cleaning and making up of the fires in the same order in which they were cleaned before the trial; (*b*) the cleaning of the ashpits (and in the case of moving bars, the flues beyond the ends of the bars). The ashes and clinkers removed from grates, ashpits and flues must be weighed, but no ashes, clinker, or fuel that may accumulate on the bars, or in the ashpits and flues after the cleaning, must be withdrawn or taken account of. During this time the principal observer should carefully watch the fires to see that they are burning evenly through and not becoming so thin as to form holes, through which air can flow, and so reduce the steam pressure. It is not necessary that they should be exactly the same thickness as at the commencement of the trial, but they ought to be of the same quality, *i.e.* as well burnt and without green coal.

"At the end of the allotted time the principal observer should again blow his whistle so that the water-feed, and if there be mechanical stokers, the coal-feed, may be stopped and the water and steam gauges read by his assistants. He himself should again gauge the thickness of the fires as at the commencement of the trial, so that the weight of fuel on the grates may be calculated. If this exceed or fall short of the weight on the grates at starting, the excess or deficiency must be deducted from or added to the weight of fuel used during the run. . . .

"The things that should be brought to the same state at the beginning and end of the trial are—

(1) The quantity of heat stored in the brickwork.
(2) The quantity of heat stored in the economiser.
(3) The steam pressure, approximately.
(4) The rate of evaporation per unit of time, since the height of the water in the gauge glass is thereby materially affected, being always higher—often 1 inch higher—when the boiler is giving off its full supply of steam than when evaporation has nearly ceased.

"In many classes of boilers the water-gauges can be read to within $\frac{1}{4}$ inch, but sometimes it is difficult to decide the level within $\frac{1}{2}$ inch. The possible algebraic sum of the errors at the beginning and end of the trial may therefore be $\frac{1}{2}$ inch to 1 inch. Hence, to obtain an accuracy of 1 per cent., the duration of the trial should be sufficient for the evaporation of a weight of water equal to 100 times the weight of a layer of water $\frac{1}{2}$ inch to 1 inch in thickness, according to the mobility of the level in the gauge glass. This will nearly always be less than the duration required to ensure an equal degree of accuracy in estimating the fuel consumption."

4. Duration of Trial.—The duration of a boiler trial is of the utmost importance, and we cannot do better than quote the following from the Committee's report:—

"When the above conditions cannot be obtained, at least approximately, a long trial is necessary to ensure accuracy, but when matters can be so arranged that the trial may be considered as a slice, so to speak, cut out of a period of uniform

working, then the duration of the trial depends principally upon the magnitude of the error likely to be made in judging the condition and thickness of the fires at the beginning and end of the trial, as compared with the weight of fuel burnt during the trial. This judgment should therefore be made by the principal observer. In most cases he will find it possible to measure the thickness of the fuel on the grate to within 1 inch. The weight of a layer of fuel 1 inch thick should therefore represent the maximum error in measuring the thickness of the fire, and this error may be in opposite senses at the beginning and end of the trial. If, therefore, A be the area in square feet of the part of the grate covered by the fuel when the measurement is made, and C the weight of a cubic foot of incandescent fuel, which will vary between 30 pounds for small slack and 20 pounds for large coal, the total error should not exceed $C \times A \times \frac{2}{12} = \frac{CA}{6}$ pounds. Therefore, if W be the number of pounds of fuel burnt per hour, the duration of the trial which will reduce the total error to 1 per cent. of the total burnt is $\frac{100CA}{6W}$ hours, and to n per cent of the total fuel burnt $\frac{100CA}{6nW}$ hours. In making use of this formula, however, it is necessary to have some regard to the quality as well as the size of the fuel. When it contains much dirt or makes a pasty clinker, the bars, if not self-cleaning, have to be cleaned at short intervals by the firemen, and at each cleaning there is loss of heat and combustible matter. The duration of the trial and the times of cleaning should therefore be so arranged as to give this loss the same average value that it would have if the trial were indefinitely prolonged. For instance, if the fuel were such as to make cleaning necessary every 4 hours, it would be unfair to make a five hours' trial; 8 hours would be the proper time; or, if it were not possible to have the trial longer than 5 hours, a more accurate result would be obtained by working for 4 hours only and cleaning the fire-grates only once.

"In connection with the duration of boiler trials, it should be remembered that if one boiler burns more coal per square

foot of grate area than another per unit of time, the trial may be of proportionally shorter duration without increase of error. The only object of prolonging the trial is to reduce the importance of an error in judgment as to the condition of the fire at the beginning and at the end of the test. This error may be taken to represent a constant weight of say w pounds of coal on a grate of given area for a trial of any length.

"When burning W pounds of coal per hour the coal burnt in N hours will be N × W, and the fractional error will be $\frac{w}{W} \times \frac{1}{N}$. The second term of this fraction is the only one usually considered, hence long trials, which are necessary with slow rates of combustion, are sometimes specified without reason in the case of trials under forced draught."

It will be seen from the above remarks that the best method of starting and stopping a trial and also the fixing of the duration of the trial are matters that require a considerable amount of experience; the principal observer should, therefore, be an engineer with special experience in this class of work. We will now consider the various methods recommended for making the necessary measurements enumerated in Art. 2.

5. Measurement of the Fuel used.—"The fuel should be weighed on a platform weighing-machine, or on beam scales, or by a *calibrated* spring balance, in parcels of 20 pounds to 200 pounds, according to the size of the boiler to be tested. It may be weighed before or during the progress of the trial. It will usually be found convenient to weigh the fuel in two boxes of equal weight and marked conspicuously No. 1 and No. 2. To prevent mistakes the Committee recommend the following method:—

"Shortly before the trial commences, No. 1 box should be filled till it balances the weights in the scale, and at the moment the trial starts its contents should be tipped out upon the stokehold floor. When the first lot is nearly finished, No. 2 box should be filled and its contents tipped out upon the floor as soon as the whole of the fuel from No. 1 box has been stoked, and so on, the boxes being used alternately. . . . The tally should be kept in the following form:—

Trial of No. ——Boiler at —— Date ——

Time of emptying No. 1 box on the stokehold floor.	Time of emptying No. 2 box on the stokehold floor.	Time interval minutes.	Remarks.
9.5	—	—	
—	9.15	10	
9.24	—	9	
—	9.36	12	

"By keeping the tally in this form the observer cannot miss or add one box; he must, if an error is made, miss or add two, and if he does this, the mistake will be detected by the inequality in the intervals of time. Inequalities in the time intervals which occur when filling up the furnaces after cleaning the grates, or from any other cause, should be noted in the 'remarks' column. The stokehold floor should be swept clean before the commencement of the trial, and any fuel lying on it at the end of the trial should be weighed and deducted from the quantity shown by the tally to have been weighed out. . . .

"Where mechanical stokers are used they should be stopped just before the commencement of the trial and the hoppers filled with unweighed fuel. At the end of the trial they should again be stopped and the hoppers filled up with weighed fuel."

6. Sampling of the Fuel, Determination of its Calorific Value and Chemical Analysis.—"When the fuel is coal, the sampling is a matter which requires great care, in order that the sample may be a good average of the coal actually used. The best way is to take half a shovelful from each box weighed out and pile it in a heap where it will neither lose nor absorb moisture. Shortly before the conclusion of the trial the heap should be well turned over and the large lumps broken up; it should then be quartered, one quarter selected for further mixing and breaking and the rest handed over to the stoker for firing. The selected portion should then be pounded, mixed, and quartered and so on until a weight of about 3 or 4 pounds remains. This final sample should be at once placed in a wide-mouthed bottle, or other air-tight receptacle, which has been carefully dried beforehand, and corked and hermetically

sealed." The calorific value is best determined by burning it with compressed oxygen in a bomb calorimeter. A chemist should analyze the sample for carbon, hydrogen, sulphur, moisture, ash, etc.

7. Measurement of the Feed Water.—As this is one of the most important measurements to be made, great care is necessary to ensure a reasonable degree of accuracy. On account of the large quantity of water evaporated during the trial of a boiler of any size it is not possible, as a rule, actually to weigh the water used; some system of volume measurement must therefore be used. The arrangement most generally

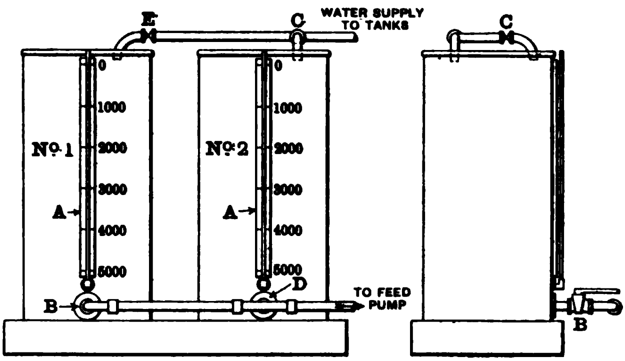


FIG. 135.—Feed-water measuring tanks.

applicable is that of two graduated measuring tanks marked No. 1 and No. 2 respectively, the contents of which may be run into a feed tank from which the water is drawn by the feed pump or injector. When this arrangement is adopted a feed tank of about one quarter the capacity of one of the measuring tanks will generally suffice. If necessary, the feed tank may be dispensed with and the measuring tanks coupled direct to the feed pump. The volume of the tanks should be such that each may hold never less than 5 minutes' supply, so that the observer may have ample time to manipulate the cocks, take the temperature of the water and book the figures. Fig. 135 shows in diagrammatic form the arrangement used by the author

when carrying out a series of tests on boilers evaporating up to 20,000 pounds of water per hour. The two measuring tanks were each fitted with a gauge glass A and accurately calibrated up to 5000 pounds at 60° F. The level in the gauge glass could be read accurately to within $\frac{1}{8}$ inch, so that assuming the error in the top and bottom readings to be in opposite directions the total maximum error possible with each tank full would be $\frac{1}{4}$ inch. The total fall in level resulting from the withdrawal of 5000 pound of water was nearly 6 feet so that the maximum possible error was negligibly small. The method of using the tanks was the same as recommended by the committee, namely:—

"Some little time before the commencement of the trial, the observer in charge of the feed measurement should fill up No. 1 measuring tank to the top level, feeding the boiler in the meantime through No. 2 tank. The feed should if possible be kept on at the rate intended for the trial until the principal observer is nearly ready to sound his whistle as a signal to begin the observations.

"As the time for starting approaches, the feed pump should be stopped, the cock D in No. 2 tank shut, and the water-level in the feed tank (if one is used) marked and recorded. After the whistle has sounded the feed pump may be again started as soon as convenient.

"The cock B on No. 1 measuring tank and the cock C for feeding No. 2 are then to be opened. While No. 1 is being emptied No. 2 should be filled. As soon as the water-level in No. 1 has fallen to the required mark, its cock B should be shut and the cock D on No. 2 opened. The cock E for No. 1 should then be opened and No. 1 filled up while No. 2 is being emptied, and so on.

"The log of the feed measurements can best be kept by entering on it the time of starting each tank, that is, the times of opening the cocks B and D as shown on next page:—

Time of commencing trial —— Date ——

Time when tanks were started.				Temperature ° F.	Time interval minutes.	Remarks.
No. 1.		No. 2.				
Hour.	Mins.	Hour.	Mins.			
9	1	—	—	58	—	
—	—	9	8	58	7	
9	16	—	—	59	8	
—	—	9	22	59	6	

"At the end of the trial the feed pump should be stopped, and whichever cock, B or D, happens to be open (say D on No. 2 tank) should be shut and the reading of its gauge glass noted. The weight of feed water used can then be easily obtained from the log-sheet. If the temperature of water in the measuring tanks differs appreciably from the temperature at which they were graduated a correction must be made in estimating the feed water supplied to the boiler. The boiler (or boilers) under test should be 'blanked off' from all others so that the measured feed water only enters the boiler under test.

8. Sampling the Ashes and Determination of the Combustible Matter therein.—The whole of the ashes and clinker collected should, after drying and weighing, be thrown into a heap and well turned over to mix them thoroughly. The heap should then be quartered and treated in exactly the same manner as the fuel (Art. 6). The sample should then be placed in an air-tight dried vessel and sent to an expert chemist for the determination of the percentage of combustible matter in it.

"It should be pointed out, however, that if a chemical analysis of the fuel has been made, it is possible to determine the quantity of combustible matter mixed with the ashes by simply deducting the weight of ash corresponding to the fuel fired as given by the fuel analysis from the weight of the material drawn from the furnaces and ashpits. In consequence of the great difficulty of obtaining a true average sample of the ash this second method is generally the more accurate. In

estimating the weight of the ash and clinker it must not be forgotten that some portion is always carried over the bridge by the draught. When the draught is strong, particularly when sprinkling stokers are in use, this proportion may be considerable, insomuch that the total weight of material drawn from the furnaces and ashpits may weigh less than the total ash contained in the fuel as determined from the chemical analysis."

9. Measurement of Temperature.—For the measurement of the air and water temperatures ordinary chemical thermometers made of hard glass are quite suitable. In measuring the feed temperature to the boiler, the best plan is to drill and tap a hole in the feed pipe close to the feed-check valve, and screw into it a steel cup into which the thermometer is inserted. The cup or pocket should be as long and as thin as possible and be filled with either mercury or cylinder oil into which the bulb of the thermometer is dipped.

Flue Temperatures.—The temperature of the flue gases should be taken as close as possible to the exit from the boiler flue. Temperatures up to 600° F. or even higher can be measured with considerable accuracy by means of mercurial thermometers made of Jena glass, with the space above the mercury filled with nitrogen or other inert gas which does not affect mercury. The thermometers used should not be too sensitive, as it is not desirable that they should be affected by every opening of the furnace doors.

"The various forms of pyrometers which work by the expansion of two different metals . . . are most of them uncertain, and cannot be relied upon to record accurately for any length of time. The most accurate and convenient method is the electrical one; either the resistance of a platinum spiral is observed, or the electromotive force generated at a thermo-electric junction is measured, and from these observations the temperature can be deduced." *

* A description of these instruments will be found in a paper "On the Measurement of High Temperatures," by Professor W. C. Roberts-Austen, C.B., F.R.S. *Proc. Inst. C.E.*, vol. cx., p. 152.

Steam Temperatures.—In reference to these measurements the Committee say:—

"These can be measured either by plunging a mercurial thermometer into a mercury cup screwed into the steam pipe, or by an electrical method.

"*Mercury Cup.*—If this method is used, great care must be taken to avoid the effects of conduction and radiation, for if the mercury cup is provided with a large solid head, the effects of radiation from this may render the readings of the thermometer quite unreliable. The mercury cup must also be of considerable length so as to project well into the pipe, and the stem of the thermometer should be immersed far enough to cover most of the mercury column. Hard glass thermometers should be used, and care must be taken to correct for change of zero.

"*Electrical Methods.*—Electrical thermometers, based either on the resistance or the thermo-electric couple method can be used for taking steam temperatures; special instruments are now made for this purpose."

10. Collection and Analysis of Flue Gases.—The sample should be taken just on the chimney side of the damper at approximately the same place at which the temperature is measured. When the boiler under test is fired by mechanical stokers, the gas samples may be drawn directly into the analysing apparatus, but when the firing is by hand, continuous collection is necessary to secure an average sample. If the gas cannot be analysed on the spot it should be collected over mercury. When the analysis can be carried out at once, collection over distilled water saturated with common salt or furnace gas yields results sufficiently accurate for practical purposes. A simple and convenient method used by the author is shown in Fig. 136. The bottle A is filled with water and connected by a rubber tube to the collecting pipe in the flue, and placed above the bottle B as shown. The pinchcock C is closed and D and E opened; water will now run from the bottle A into the bottle B, the rate of flow being regulated by the opening of the pinchcock D. As water runs out of A, the gas will be drawn into it. The sample thus drawn into the bottle A will not be a true sample of the flue gases because it

will contain air which was present in the collecting tube F. The pinchcocks D and E are now closed, the bottles A and B interchanged, and cocks C and D opened; the water in B will now flow back into A, driving the mixture of gas and air out of A through the cock C. When the bottle A is again full of water, C is closed and the bottles interchanged again, bringing A to the top again. The cocks D and E are now reopened, and the true sample of gas drawn into A as before. When the bottle A is full of gas, E should be closed, and the bottles again interchanged. The gas in A will now be under pressure due

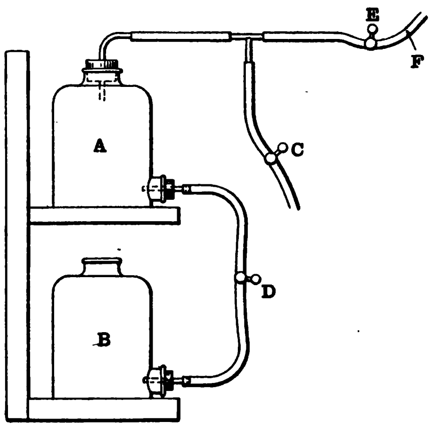


FIG. 136.—Gas sampling bottles.

to the head of water above it, and if there is any slight leakage at the cork air will not leak in, but only a little gas will leak out. The sample will thus not be spoiled, and in addition will be under pressure to drive it rapidly into the analysing apparatus.

"The most convenient apparatus for analysing the gas on the spot is the Orsat apparatus (Fig. 137) as it requires no supply of pure water and no bottles of chemicals." G is a eudiometer surrounded by a glass tube which is filled with water to form a jacket, and thus ensure a uniform temperature for all the gas measurements. H, K and L are flasks containing

solutions of caustic soda, pyrogallic acid, and cuprous chloride respectively, for the absorption of carbon dioxide (CO_2), oxygen (O), and carbon monoxide (CO). These solutions are made as follows:—

For CO_2 (in flask H).—One part of caustic soda (NaOH) to two of water by weight. Caustic potash (KOH) may be used instead of caustic soda if desired.

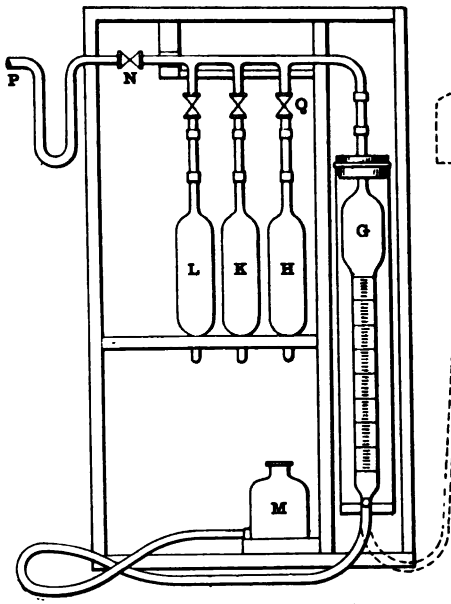


FIG. 137.—Orsat apparatus.

For Oxygen (in flask K).—One part by weight of pyrogallic acid dissolved in 3 parts of water, and 24 parts of either caustic soda or caustic potash dissolved in 16 parts of water. Sticks of phosphorus may be used instead of this alkaline pyrogallic if desired.

For CO in flask L.—A solution of cuprous chloride in hydrochloric acid, made by dissolving copper oxide (CuO) in about 20 times its weight of strong hydrochloric acid, and allowing it

to stand in a rubber corked flask containing copper wire until the solution becomes colourless.

These reagent flasks contain a number of glass tubes packed together and open at the ends; by this means a large surface of reagent is exposed to the gas as it is driven over into the flask. Duplicate flasks are arranged behind the reagent flasks H, K, L, for the reception of the reagents as they are expelled from those flasks with which each duplicate flask is respectively connected. M is an aspirator, and at N is a three-way cock making a straight-through connection and also a connection with the atmosphere.

The method of analysing the sample of gas is as follows:— The bottle A (Fig. 136) containing the gas is connected at P (Fig. 137). All the other cocks being closed, N is opened to the atmosphere and C or E (Fig. 136) opened, and by raising the aspirator M the eudiometer and the capillary tube are filled with the liquid contained in the aspirator. The cock N is now turned so as to make a through connection, the aspirator is lowered, and 100 cubic centimetres of gas are drawn into the eudiometer. When the measurement is being made the level of liquid in the aspirator and eudiometer must be the same. As soon as the sample is obtained the cock N is closed and the gas analysed in the following manner.

The percentage of carbon dioxide is first obtained by opening the cock Q and raising M, so driving the gas out of the eudiometer into the flask H which contains the caustic soda. On drawing back the gas, the cock Q is closed as soon as the reagent in H rises to its original level. This operation is repeated several times, observing each time the decrease in volume of the gas. When no further decrease in volume is observed, all the CO_2 will have been absorbed and the total decrease gives the percentage of CO_2 present in the sample.

The oxygen is next absorbed by passing the gas into the flask K containing the pyrogallic or sticks of phosphorus, the operation being carried out in the same way as for CO_2. When all the oxygen is removed, the flask L is used in exactly the same way and the percentage of CO measured.

"With this apparatus it is not possible to determine

accurately the amount of CO present in the gas, because it is seldom present in such quantity as to make its determination trustworthy. The amount of CO_2 and oxygen present can be determined to within 0·5 per cent., and if it is found when these two constituents are entirely removed that the residual nitrogen is 81 or 82 per cent. of the whole, then it may be conjectured that CO is present. If it is suspected that CO is present, then samples of the gas should be submitted to a chemist skilled in gas analysis to determine the proportion present."

11. Measurement of the Dryness Fraction of the Steam.—The most usual method of measuring the dryness of the steam is by the use of a wire-drawing or throttling calorimeter. The calorimeter is fitted up as shown in Fig. 138, and to reduce radiation losses the calorimeter supply pipe, between the calorimeter and the main steam pipe, should be as short as possible and together with the calorimeter itself should be well lagged with a non-conductor of heat, such as hair felt or asbestos rope. The action of this calorimeter depends upon the fact that the total heat of saturated steam at high pressure is greater than at low pressure, and when the pressure falls, the excess of heat is liberated, and goes first to evaporate any moisture present in the steam, and then, if the heat is sufficient, to superheat the steam at the lower pressure in the calorimeter. The steam is allowed to pass from the main steam pipe through the small orifice in the wall of the calorimeter, where the pressure falls, and away by the exhaust pipe. The temperature of the steam in the calorimeter after expansion is taken by a thermometer inserted in a pocket filled with cylinder oil, while the pressure of steam in the calorimeter is shown on the mercury manometer or column gauge. If there is so much moisture in the steam that the heat liberated is insufficient to evaporate it and *then* to superheat it, this calorimeter would not be applicable. The throttling calorimeter cannot be relied upon if the steam contains more than about 2·5 per cent. of moisture.

In order to use the instrument, the globe valve is opened and steam allowed to flow freely through the instrument until the thermometer registers its maximum reading (this will

usually take about 10 minutes). The thermometer reading is then taken together with the absolute pressure in the main steam pipe and the reading of the manometer. Then by equating the heat in the steam before expansion to that after expansion, the dryness fraction of the steam can be calculated. An example will make this clear.

Example.—In a test with the throttling calorimeter, the following readings were taken :—Barometer reading, 29·9

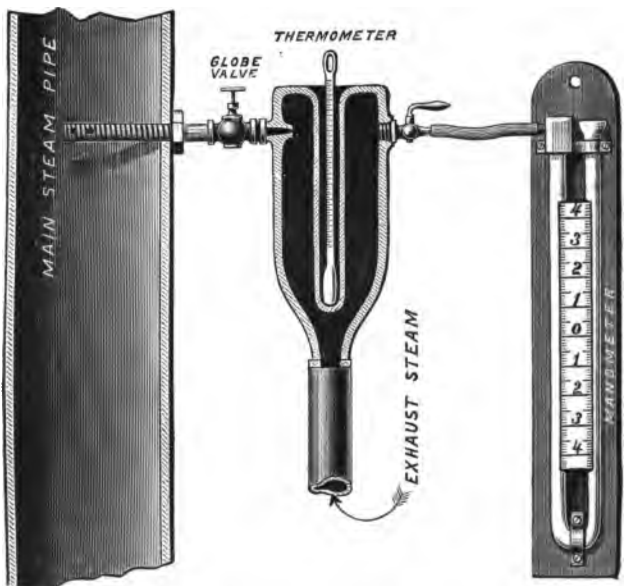


FIG. 138.—Throttling calorimeter.

inches; pressure in main steam pipe, 80·3 pounds per square inch gauge; temperature after throttling, 261° F.; pressure after throttling (above atmospheric, *i.e.* manometer reading), 4·2 inches. Estimate the dryness fraction of the steam in the main steam pipe.

Absolute pressure in main steam pipe

$$= \text{gauge pressure} + \text{atmospheric}$$
$$= 80{\cdot}3 + 14{\cdot}7$$
$$= 95 \text{ pounds per square inch.}$$

Absolute pressure after throttling

$$= \text{gauge pressure} + \text{atmospheric}$$
$$= 4{\cdot}2 + 29{\cdot}9$$
$$= 34{\cdot}1 \text{ inches of mercury}$$
$$= 34{\cdot}1 \times 0{\cdot}49 = 16{\cdot}7 \text{ pounds per square inch.}$$

Heat per pound before expansion

= heat per pound after expansion.

$$h_1 + x\mathrm{L}_1 = \mathrm{H}_2 + 0{\cdot}48(\mathrm{T} - t_2)$$

where h_1 = sensible heat before expansion at 95 pounds absolute,
L_1 = latent heat " " " "
x = dryness fraction required,
H_2 = total heat of dry and saturated steam after expansion at 16·7 pounds absolute,
t_2 = temperature of saturation at 16·7 pounds absolute,
T = temperature after expansion.

Substituting the above from steam tables, we have

$$h_1 + x\mathrm{L}_1 = \mathrm{H}_2 + 0{\cdot}48(\mathrm{T} - t_2)$$
$$294{\cdot}5 + x \times 890{\cdot}7 = 1152{\cdot}5 + 0{\cdot}48(261 - 220)$$
$$= 1172{\cdot}2$$
$$\therefore \quad x = \frac{1172{\cdot}2 - 294{\cdot}5}{890{\cdot}7} = 0{\cdot}985.$$

Hence the steam is 98·5 per cent. dry, or it contains 1·5 per cent. of moisture. When the steam is too wet to use a throttling calorimeter some form of separating calorimeter may be used. Fig. 139 shows an improved form of separating calorimeter by Messrs. Schäffer & Budenberg. In this instrument, the moisture contained in the steam is actually separated from the steam, and the degree of moisture is arrived at by comparing the quantity of water separated with the total quantity of steam from which it has been obtained. Referring to Fig. 139, it will be seen that the instrument consists of two vessels, one being interior to the other; the outer vessel surrounds the inner one so as to leave a space which answers for a steam jacket, the interior vessel is provided with a water gauge 10 and a graduated scale 12. The steam whose quality is to be determined is supplied through the pipe 6 into the upper

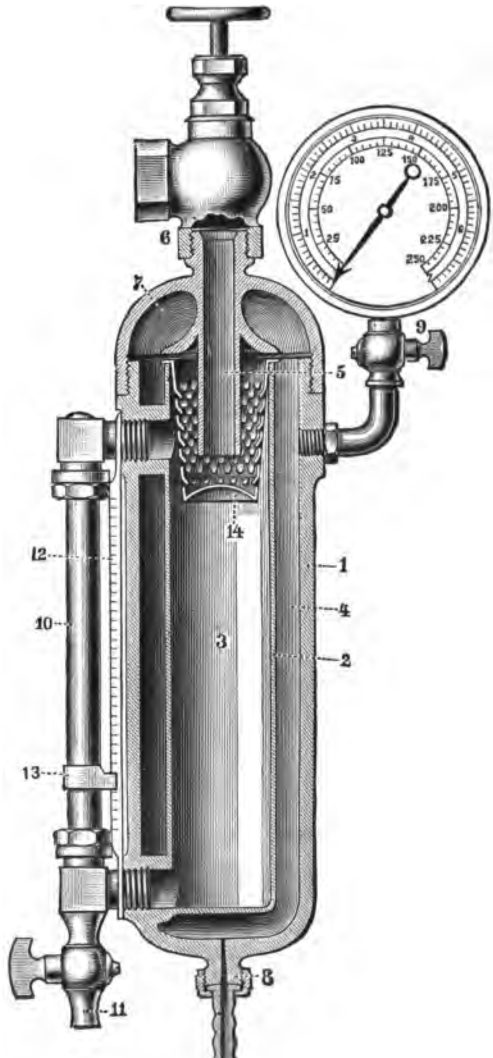


Fig. 139.—Improved separating calorimeter.

portion of the interior vessel. The water in the steam is thrown downward into the cup 14, together with more or less of the steam; the course of the steam and water is then changed through an angle of nearly 180°, which causes the greater weight of water to be thrown outwards through the meshes in the cup into the space 3 below. The cup serves to prevent the current of steam from taking up any moisture which has already been thrown out by inertia. The meshes or fins project upwards into the inside of the cup, so that any water intercepted will drip into the inner chamber 3; the steam then passes upwards and enters through a narrow passage into the top of the chamber 4, which forms a steam jacket surrounding the chamber 3. The steam is discharged from the chamber 4 through an orifice 8, of known area, which is much smaller than any section of the passages through the calorimeter, so that the steam in the chamber 4 suffers no sensible reduction in pressure by passing through the calorimeter. The velocity of the steam in the interior chamber 3 is very small to allow the moisture to settle down.

The quantity of water thus collected is measured by means of the gauge glass 10 and the pointer 13 on the scale 12. This scale is graduated in pounds and hundredths by actual calibration with water at a temperature of 100° F. The quantity of dry steam passing through the instrument was, in the earlier forms of the instrument, measured by passing the steam away into a condensing vessel containing water, its weight being determined from the increase in weight of the water in the condensing vessel. In the latest form of the instrument, a pressure gauge 9 is attached to the steam jacket of the calorimeter, and in addition to the ordinary pressure scale the dial has a second outer scale graduated by trial and showing the weight of steam discharged at every pressure in ten minutes.

The instrument should be connected to the main steam pipe and lagged with hair felt to prevent loss by radiation, as described above for the throttling calorimeter. To use the instrument, blow steam through before making observations until it is thoroughly heated. Then blow steam through for exactly ten minutes, keeping the pressure in the steam jacket

as nearly constant as possible. Note the readings on the scale 12 at the beginning and end of the test, and also note the average position of the pointer on the gauge dial during the test. The percentage of moisture is then found by dividing the weight of separated water, as shown on the water gauge 12, by the sum of this quantity and that shown on the outer scale of the gauge dial.

Whatever form of instrument is used in order to determine the dryness of the steam, the chief obstacle to contend with is the difficulty experienced in taking a representative sample of the steam from the main steam pipe. It should be remembered that the steam in the main steam pipe is not at rest; on the contrary, it is moving along the pipe with a high velocity. If, therefore, the calorimeter sampling pipe, which is screwed into the main steam pipe, consists of a simple pipe with an open end, it will act as a fairly good water separator, because the steam in the main steam pipe will be sweeping across the open end at a high velocity, and in order to enter the sampling pipe will have to turn suddenly through a right angle; by so doing, a large proportion of the moisture it contains will be thrown off, due to its greater inertia, and the steam taken to the calorimeter will be drier than that in the main steam pipe. For this reason it is desirable that the sampling pipe should extend about three-quarters across the diameter of the main steam pipe, and two or more holes, about $\frac{1}{4}$ inch in diameter, should be drilled through it at different points along its length. By this means the probability is that a fairly good representative sample of the steam in the main steam pipe will be obtained.

12. Method of Working out the Heat Account and Deductions for a Boiler Trial.—An actual example will make this clear. The data obtained from a trial on two Babcock and Wilcox boilers, working in conjunction with a Green's economiser, together with the heat account and deductions, is shown on pages 382–385 on the standard schedules recommended by the Committee of the Institution of Civil Engineers on Steam Boiler Trials.

BOILER. Sheet I. GENERAL DESCRIPTION AND DIMENSIONS.

Type of Boiler. *Water Tube.* Made by *Messrs. Babcock & Wilcox.*
Maker's rating of the output of the Boilers *10,000* lbs. of steam per hour.
Test made at an output of *9900* lbs. of steam per hour.
Object of the Trial. *Contract conditions.*

Reference number.	GENERAL DESCRIPTION OF BOILER AND LEADING DIMENSIONS.
1	*Ordinary Babcock & Wilcox type.*
2	Method of starting and stopping the test. *The trial was started and finished by the third method described on page 362 (or by the method C described on p. 37 of the Committee's Report).*
3	Method of stoking and average thickness of fire. *Hand firing.*
4	Production of draught. *Natural.*
5	Chimney, height. *Above grate 146* ft. Area at bottom, *23·6* sq. ft.; top, *23·6* sq. ft.
6	Total grate surface (excluding dead plate) sq. ft. *58 (2 boilers)*
7	Grate area occupied by air spaces between bars . . . . sq. ft. *16*
8	Total effective heating surface . . . . sq. ft. *3238 (2 boilers)*
9	Capacity of water space . . } at ______ inches in gauge glass { c. ft.______
10	,, steam space . . } at ______ inches in gauge glass { c. ft.______
11	Area of water surface in boiler } at ______ inches in gauge glass { sq. ft.______

BOILER. Sheet II. DATA DEDUCED FROM OBSERVATIONS.

Reference number.	Particulars of observations.	Abstract of observations.	Remarks.
12	Duration of trial from —— to —— . . . . hours	*7·5*	
	FUEL.		
13	Short description	*Good gas coke*	*From Manchester Corporation Gas Works.*
14	Fired per hour . . . . lbs.	*1152*	
15	Analysis by weight of dried fuel—Carbon . per cent.	*87·30*	
	" " " Hydrogen per cent.	*0·78*	
	" " " Sulphur . per cent.	—	
	" " " Ash . . per cent.	*8·27*	
	" " " Oxygen and other matters per cent.	*2·36*	
16	Moisture in fuel as fired . . . . per lb.	*0·029*	
17	Calorific value of dried fuel ("lower" value) B.Th.U. . . . . per lb.	*12830*	*Bomb calorimeter.*
18	Carbon value per lb. of dried fuel . . . . lb.	*0·88*	
	ASH AND CLINKER.		
19	Total per hour . . . . lbs.	*95*	
20	Carbonaceous matter in ash per hour . . . . lbs.	*nil*	
	FLUE GASES.	*By vol.* / *By weight.*	
21	Analysis of dry flue gases—Carbonic acid . per cent.	*9·88* / *14·50*	
	" " " Carbonic oxide per cent.	*0·05* / *0·05*	
	" " " Oxygen . . . per cent.	*9·82* / *10·60*	
	" " " Nitrogen (by difference) . . per cent.	*80·25* / *74·90*	
22	Average temperature leaving boiler flues . . . °F.	*561·8*	
23	Mean specific heat of products of combustion B.Th.U.	*0·237*	
	AIR AND DRAUGHT.		
24	Temperature of outside air . . . . °F.	*63·2*	
25	Barometric pressure (——ins. mercury) lbs. per sq. in.	*14·5*	
26	Pressure in ash pit (if forced air supply) ins. of water	*Not taken*	
27	" over fire (" ") ins. of water	*Not taken*	
28	Draught at gas exit from boiler . . . ins. of water	*½ to 7/16*	
29	" base of chimney . . . . ins. of water	*Not taken*	
30	Weight of steam per hour used in producing draught lbs.	—	
	FEED WATER.		
31	From pump, economiser or feed heater per hour lbs.	*9896*	
32	Temperature of feed to boiler . . . . °F.	*190*	
	STEAM.		
33	Gauge pressure . . . . lbs. per sq. in.	*143·8*	
34	Abolute pressure . . . . lbs. per sq. in.	*158·3*	*By throttling calorimeter.*
35	Total moisture in steam . . . . per lb.	*0·01*	
36	Temperature of saturation . . . . °F.	*362·4*	

BOILER. Sheet III. HEAT ACCOUNT AND DEDUCTIONS.

Reference number.	Heat account (per lb. of dried fuel).	B.Th.U.	Per cent.
37	Total heat value of 1 lb. of dried fuel . . .	*12,830*	*100·0*
38	Heat transferred to water (*thermal efficiency*) .	*9081*	*70·74*
39	Heat carried away by products of combustion	*1332*	*10·39*
40	Heat carried away by excess air	*1304*	*10·16*
41	Heat lost in evaporating and in superheating moisture mixed with fuel	*37*	*0·28*
42	Heat lost by incomplete combustion	*44*	*0·34*
43	Heat lost by unburnt carbon in ash	nil	nil
44	Balance of heat account: Errors of observation, and unmeasured losses, such as—those due to radiation, escape of unburnt hydrocarbons, superheating moisture in air, loss in hot ashes, etc.	*1032*	*8·09*
	Total of lines 38 to 44, equal to line 37 .	*12,830*	*100·00*
	DEDUCTIONS.		
45	Heat transmitted per square foot of heating surface per hour	B.Th.U.	*31·35*
46	Weight of fuel fired per square foot of grate area per hour	lbs.	*19·86*
47	Weight of dried fuel fired per square foot of grate area per hour	lbs.	*19·28*
48	Water evaporated per lb. of fuel as fired . .	lbs.	*8·59*
49	Equivalent evaporation from and at 212° F. per lb. of fuel as fired	lbs.	*9·12*
50	Water evaporated per lb. of dried fuel . . .	lbs.	*8·84*
51	Equivalent evaporation from and at 212° F. per lb. of dried fuel	lbs.	*9·39*
53	Weight of feed, from and at 212° F., per square foot of heating surface per hour	lbs.	*3·29*
54	Velocity of steam across water surface . . .	feet per second	
55	Air used per lb. of dried fuel	lbs.	*21·38*
56	Ratio of air used to air theoretically needed		*2·05*

ECONOMISER. Sheet I. General Description and Dimensions.

Reference number.	Economiser. (Heating of feed by flue gases.)
57	General description of Economiser, *Made by Messrs. Green, of Wakefield. Total number of pipes 96, in 12 lines of 8 pipes each.*
58	Heating surface of economiser . . . . *960* sq. feet.

Data deduced from Observations.

Reference number.		Abstract of observations.		Remarks.
61	Weight of feed water entering economiser per hour . . lbs.	*9917*		Economiser leak was 21 lbs. per hour.
62	Temperature of feed into . °F.	*105·1*		
63	" " out of °F.	*190*		
64	" of flue gases into °F.	*561·8*		
65	" " " out of °F.	*361·4*		
66	Analysis of dry flue gases leaving economiser—	*By volume.*	*By weight.*	
66	Carbonic acid . . . per cent.	*8 79*	*12·90*	
66	Carbonic oxide . . per cent.	*0·05*	*0·05*	
66	Oxygen . . . . . . per cent.	*10·89*	*11·55*	
66	Nitrogen (by difference) per cent.	*80·27*	*75·50*	
67	Mean specific heat of flue gases leaving economiser . B.Th.U.	*0·238*		

ECONOMISER. Heat Account and Deductions.

Reference number.	Heat account (per lb. of dried fuel).	B.Th.U.	Per cent.
77	Heat received from boiler flues per lb. of dried fuel (reckoned from air temperature)	*2636*	*100·0*
78	Heat transferred to water (*efficiency of economiser*)	*750*	*28·22*
79	Heat carried off in chimney gases . . .	*1755*	*66·58*
80	Balance of heat account, including errors of observation and difference of heat contained in brickwork at beginning and end of test, etc.	*131*	*5·20*
	Total of lines 78 to 80, equal to line 77 .	*2636*	*100·0*
85	Heat transmitted per square foot of heating surface of economiser per hour . .	B.Th.U.	*873*
87	Thermal efficiency of boiler and economiser combined	per cent.	*76·53*

The method of working out much of the above data is self-evident and requires no explanation. In what follows only the most important items are worked out fully.

Line 23. *Mean Specific Heat of Products of Combustion.*—In order to estimate this item we must first find the actual weight of CO_2, O_2, H_2O, and N_2 present in the products of combustion from 1 pound of the dried fuel. The sum of these four quantities will obviously be the total weight of the products of combustion.

The total weight of flue gases per pound of fuel is first found as follows:—

Using equation 3, Art. 8, Chap. II., we have

$$\left.\begin{array}{l}\text{Air supplied per pound}\\ \text{of dried fuel}\end{array}\right\} = \frac{N_2}{33(CO_2 + CO)} \times C$$

$$= \frac{80 \cdot 25}{33(9 \cdot 88 + 0 \cdot 05)} \times 87 \cdot 3$$

$= 21 \cdot 38$ pounds, which is the item to fill in for Line 55.

The total weight of gases $= 21 \cdot 38 +$ combustible in 1 pound of fuel

$$= 21 \cdot 38 + 0 \cdot 873 + 0 \cdot 0078$$

$$= 22 \cdot 26 \text{ pounds.}$$

$$\left.\begin{array}{l}\text{Minimum quantity of air}\\ \text{theoretically required}\\ \text{per pound of fuel}\end{array}\right\} = 11 \cdot 6C + 34 \cdot 8H \qquad (4)\text{ p. } 31$$

$$= 11 \cdot 6 \times 0 \cdot 873 + 34 \cdot 8 \times 0 \cdot 0078$$

$$= 10 \cdot 12 + 0 \cdot 27$$

$$= 10 \cdot 39 \text{ pounds.}$$

Excess air supplied = air actually supplied − minimum theoretical quantity

$$= 21 \cdot 38 - 10 \cdot 39$$

$$= 10 \cdot 99 \text{ pounds.}$$

$$\therefore \left.\begin{array}{l}\text{Weight of products}\\ \text{of combustion from}\\ \text{1 pound of fuel}\end{array}\right\} = 22 \cdot 26 - 10 \cdot 99$$

$$= 11 \cdot 27 \text{ pounds.}$$

We now require the weight of each constituent in this 11·27 pounds of products of combustion. Following the method of Art. 10, Chap. II., we first convert the volumetric analysis of the flue gases into the analysis by weight:—

Constituent.	By volume. × Relative weight. =		By weight.		
CO_2	$= 0·0988 \times 22 =$	2·174	= 0·145	= 14·5	per cent.
CO	$= 0·0005 \times 14 =$	0·007	= 0·0005	= 0·05	"
O_2	$= 0·0982 \times 16 =$	1·571	= 0·1055	= 10·55	"
N_2	$= 0·8025 \times 14 =$	11·235	= 0·749	= 74·9	"
	Total . .	14·987	1·000	100·0	

We next require the proportion of carbon burned to CO_2 and CO respectively. By the method on p. 47, we have

In 14·5 parts of CO_2 there are $14·5 \times \frac{3}{11} = 3·95$ parts of carbon.
" 0·05 " " CO " " $0·05 \times \frac{3}{7} = 0·02$ " " "
Total $= 3·97$ " " "

$$\therefore \text{ Proportion of carbon burned to } CO_2 = \frac{3·95}{3·97}$$

$$\text{" \quad " \quad " } \quad CO = \frac{0·02}{3·97}$$

Hence per pound of dried fuel burned, we have

$$\text{Weight of } CO_2 = \tfrac{11}{3} \times 0·873 \times \frac{3·95}{3·97} = 3·20 \text{ pounds.}$$

$$\text{Weight of } CO = \tfrac{7}{3} \times 0·873 \times \frac{0·02}{3·97} = 0·01 \text{ pound.}$$

$$\begin{aligned}\text{Weight of steam formed} &= \text{weight of hydrogen} \times 9 \\ &= 0·0078 \times 9 = 0·070 \text{ pound.}\end{aligned}$$

$$\begin{aligned}\text{Weight of nitrogen (by difference)} &= 11·27 - (3·20 + 0·01 + 0·07) \\ &= 11·27 - 3·28 \\ &= 7·99 \text{ pounds.}\end{aligned}$$

We next require the proportion by weight of each constituent in the products of combustion.

$$CO_2 = \frac{3 \cdot 20}{11 \cdot 27} = 0 \cdot 2839$$

$$CO = \frac{0 \cdot 01}{11 \cdot 27} = 0 \cdot 0008$$

$$H_2O = \frac{0 \cdot 07}{11 \cdot 27} = 0 \cdot 0062$$

$$N_2 = \frac{7 \cdot 99}{11 \cdot 27} = 0 \cdot 7091$$

$$\text{Total} \quad 1 \cdot 0000$$

Taking the following values of the specific heats of these constituents

$$CO_2 = 0 \cdot 216 \qquad H_2O = 0 \cdot 48$$
$$CO = 0 \cdot 245 \qquad N_2 = 0 \cdot 244$$

we have (p. 37)

Mean specific heat of the products of combustion

$$= (0 \cdot 2839 \times 0 \cdot 216) + (0 \cdot 0008 \times 0 \cdot 245) + (0 \cdot 0062 \times 0 \cdot 48) + 0 \cdot 7091 \times 0 \cdot 244 = 0 \cdot 237$$

Line 38.—This item will be the heat of formation per pound of steam, multiplied by the weight of feed water per pound of dried fuel.

Heat of formation or the heat supplied by the boiler per pound of steam = increase in sensible heat + x × Latent heat (Art. 9, Chap. V.).

From steam tables (pp. 406–407) we find that at 158·3 pounds per square inch absolute, the sensible heat is 334·5 B.Th.U. and the latent heat 859·5 B.Th.U. From Line 35 the dryness fraction x is 0·99. Hence

$$\text{Heat of formation} = 334 \cdot 5 - (190 - 32) + 0 \cdot 99 \times 859 \cdot 5$$
$$= 176 \cdot 5 + 850$$
$$= 1026 \cdot 5 \text{ B.Th.U.}$$

This quantity might also be calculated from the expression

$$H_1 - h_0 - w(H_1 - h_1)$$

where H_1 = total heat of evaporation at boiler pressure (158·3 pounds absolute),

h_1 = sensible heat at boiler pressure,

h_0 = sensible heat at feed temperature,

w = weight of moisture in 1 pound of steam = 0·01 pound.

From steam tables we find:—$H_1 = 1194\cdot0$ B.Th.U.; $h_1 = 334\cdot5$ B.Th.U.; $h_0 = 159$ B.Th.U. Hence

$$\begin{aligned}\text{Heat of formation} &= (1194\cdot0 - 159) - 0\cdot01(1194\cdot0 - 334\cdot5)\\ &= 1035\cdot0 - 8\cdot5\\ &= 1026\cdot5 \text{ B.Th.U. as above.}\end{aligned}$$

$$\text{Now water per pound of dried fuel} = \frac{\text{feed water per hour}}{\text{dried fuel per hour}}$$

$$= \frac{9896 \text{ (line 31)}}{1152 - 0\cdot029 \times 1152 \text{ (lines 14 and 16)}}$$

$$= \frac{9896}{1118\cdot6} = 8\cdot846 \text{ pounds.}$$

Hence the amount of heat transferred to the water per pound of dried fuel will be

$$1026 \times 8\cdot846 = 9081 \text{ B.Th.U.}$$

Line 39.—The heat carried away by the products of combustion will be

$$\begin{aligned}&\text{weight of products} \times \text{specific heat}\\ &\qquad \times \Big(\underset{\text{Line 22}}{\text{temperature of products}} - \underset{\text{Line 24}}{\text{air temperature}}\Big)\\ &\qquad = 11\cdot27 \times 0\cdot237 \times (561\cdot8 - 63\cdot2)\\ &\qquad = 1332 \text{ B.Th.U.}\end{aligned}$$

Line 40.—The heat carried away by the excess air will be

$$\begin{aligned}&\text{Weight of excess air} \times \text{specific heat of air} \times (\text{Line 22} - \text{Line 24})\\ &\qquad = 10\cdot99 \times 0\cdot238 \times (561\cdot8 - 63\cdot2)\\ &\qquad = 1304 \text{ B.Th.U.}\end{aligned}$$

Line 41.—Assuming that the moisture in the fuel as fired is at the temperature of the air, viz. 63·2° F., this item will be the quantity of heat required to raise the temperature of the moisture from 63·2° F. to 212° F., then to evaporate it at this temperature, and finally to superheat it to the temperature of the flue gases, viz. 561·8° F.

Hence, heat lost in evaporating and in superheating moisture mixed with fuel is

$$\begin{aligned}&0\cdot029[(212 - 63\cdot2) + 966 + 0\cdot48(561\cdot8 - 212)]\\ &= 0\cdot029[148\cdot8 + 966 + 167\cdot9]\\ &= 0\cdot029 \times 1282\cdot7\\ &= 37 \text{ B.Th.U.}\end{aligned}$$

Line 42.—From the calculations already made for Line 23, p. 387, we see that the proportion of carbon in the fuel which is burned to CO is $\frac{0 \cdot 02}{3 \cdot 97}$.

$$\left.\begin{array}{l}\text{Hence, heat lost by incom-}\\ \quad\text{plete combustion by p. 47}\end{array}\right\} = 0 \cdot 873 \times \frac{0 \cdot 02}{3 \cdot 97} \times (14{,}500 - 4400)$$
$$= 44 \text{ B.Th.U.}$$

where 14,540 is the calorific value of carbon when completely burned to CO_2 and 4400 is the calorific value of carbon when incompletely burned to CO.

Line 45.—The heat transmitted per square foot of heating surface per hour will be

$$\frac{\text{Heat transferred to water (Line 38)} \times \text{lbs. of dried fuel per hour}}{\text{Heating surface (Line 8)}}$$
$$= \frac{9081 \times 1118 \cdot 6}{3238} = 3135 \text{ B.Th.U.}$$

Line 67.—The method of calculating the mean specific heat of the flue gases has already been fully explained in Art. 10, Chap. II. The necessary calculations are as follows:—

$$CO_2 = 0 \cdot 0879 \times 22 = 1 \cdot 934 = \frac{1 \cdot 934}{14 \cdot 92} = 0 \cdot 129 \text{ proportion by weight.}$$
$$CO = 0 \cdot 0005 \times 14 = 0 \cdot 007 = \frac{0 \cdot 007}{14 \cdot 92} = 0 \cdot 0005 \quad ,, \quad ,,$$
$$O_2 = 0 \cdot 1089 \times 16 = 1 \cdot 742 = \frac{1 \cdot 742}{14 \cdot 92} = 0 \cdot 1155 \quad ,, \quad ,,$$
$$N_2 = 0 \cdot 8027 \times 14 = 11 \cdot 237 = \frac{11 \cdot 237}{14 \cdot 92} = 0 \cdot 755 \quad ,, \quad ,,$$
$$\text{Total} \quad . \quad . \quad 14 \cdot 920 \qquad\qquad 1 \cdot 000$$

Therefore in 100 pounds of dry flue gases there are

$$12 \cdot 9 \times \tfrac{12}{44} + \cdot 05 \times \tfrac{12}{28}$$
$$= 3 \cdot 52 + 0 \cdot 02$$
$$= 3 \cdot 54 \text{ pounds of carbon.}$$

Hence the weight of dry flue gases per pound of dried fuel will be

$$\frac{100 \times 0 \cdot 873}{3 \cdot 54} = 24 \cdot 66 \text{ pounds.}$$

But the 0·0078 pound of hydrogen in the fuel produces $0{\cdot}0078 \times 9 = 0{\cdot}070$ pound of steam, hence the actual weight of each constituent in the flue gases from 1 pound of dry fuel is

Constituent.	Weight in pounds per pound of dry fuel burned.	Weight in pounds of each constituent in 1 pound of flue gases.
CO_2	$24{\cdot}66 \times 0{\cdot}129 = 3{\cdot}181$	$\frac{3{\cdot}181}{24{\cdot}729} = 0{\cdot}1286$
CO	$24{\cdot}66 \times 0{\cdot}0005 = 0{\cdot}012$	$\frac{0{\cdot}012}{24{\cdot}729} = 0{\cdot}0005$
O_2	$24{\cdot}66 \times 0{\cdot}1155 = 2{\cdot}848$	$\frac{2{\cdot}848}{24{\cdot}729} = 0{\cdot}1151$
N_2	$24{\cdot}66 \times 0{\cdot}7550 = 18{\cdot}618$	$\frac{18{\cdot}618}{24{\cdot}729} = 0{\cdot}7528$
H_2O	0·070	$\frac{0{\cdot}07}{24{\cdot}729} = 0{\cdot}0030$
	24·729	1·0000

Hence the mean specific heat of the flue gases is

$$(0{\cdot}1286 \times 0{\cdot}216)+(0{\cdot}0005 \times 0{\cdot}245)+(0{\cdot}1151 \times 0{\cdot}218)$$
$$+(0{\cdot}7528 \times 0{\cdot}244)+(0{\cdot}003 \times 0{\cdot}48)=0{\cdot}238$$

Line 77.—This item is the sum of lines 39 and 40 of Boiler Sheet No. III., in this case 1332 + 1304 = 2636 B.Th.U.

Line 78.—This line is the weight of feed water per pound of dried fuel (line 50) multiplied by the rise of temperature of the water in passing through the economiser, *i.e.*

$$8{\cdot}84 \times (190 - 105{\cdot}1) = 750 \text{ B.Th.U.}$$

Line 79.—The method of calculating this item is exactly the same as for lines 39 and 40. The heat carried off in the chimney gases will be

$$24{\cdot}729 \times 0{\cdot}238 \times (361{\cdot}4 - 63{\cdot}2)$$
$$= 24{\cdot}729 \times 0{\cdot}238 \times 298{\cdot}2$$
$$= 1755 \text{ B.Th.U.}$$

Line 85.—This item is equal to

$$\frac{\text{weight of dried fuel burnt per hour} \times \text{heat transferred to water (line 78)}}{\text{heating surface of economiser}}$$

$$= \frac{1118 \cdot 6 \times 750}{960}$$

$$= 873 \text{ B.Th.U.}$$

Line 87.—The thermal efficiency of boiler and economiser combined will obviously be :—

$$\frac{\text{Total amount of heat passed into the feed water per pound of dried fuel}}{\text{Calorific value of dried fuel}}$$

$$= \frac{\text{line 38} + \text{line 78}}{\text{line 17}}$$

$$= \frac{9081 + 750}{12{,}830} = 0 \cdot 7653 \text{ or } 76 \cdot 53 \text{ per cent.}$$

13. Tests of Large Boilers at Detroit.*—The following particulars are of very great interest inasmuch as these boilers appear to be the largest ever constructed. Three of these boilers have been in regular service for 18 months, and two more are now in course of erection in the works of the Detroit Edison Co. Two boilers were subjected to the most careful and exhaustive tests, one being fitted with Roney underfeed stokers, and the other with Taylor underfeed stokers.

The boilers (Figs. 140 and 140A) were built by the Babcock & Wilcox Co. Each contained 23,654 sq. feet of heating surface, and was provided with superheaters for supplying approximately 150° F. of superheat. The grate area was 446 square feet for the Roney stokers and 405 square feet for the Taylor stokers. The tests were run under the direct supervision of Mr. D. S. Jacobus, assisted by men chosen from the staff of the Babcock & Wilcox Co. In addition to men furnished by the Detroit Edison Co., the Solway Process Co. provided a number of observers and made analyses and tests of the calorific values of the coal used in each test. Duplicate samples were taken and

* D. S. Jacobus, *Proc. Am. Soc. M.E.*, 1911.

the work was done a second time in the laboratory of the Babcock & Wilcox Co. The average of the results secured by the two laboratories was used in working up the tests.

All the feed water was weighed in tanks each of which had a capacity of 20,000 pounds. Three such tanks and weighing machines were provided, two being used regularly and the third held for a space in case any irregularity developed in either of the other two. The coal was weighed by means of four special scales carrying overhead hoppers, each of about 2500 pounds capacity. Readings of temperature, pressures, etc., were taken every half-hour and every precaution taken to ensure as high a degree of accuracy as possible. The analyses of the gases after leaving the boiler dampers were made with an Orsat apparatus, the samples of gas being taken from the middle of the flues.

Results of the Tests.—A résumé of the results obtained is given in the accompanying table. It will be seen that the combined efficiency of the boiler and furnace varies from about 80 per cent. at slightly below rating to about 76 per cent. at double rating. In obtaining these efficiencies the steam used for driving the stokers and producing the forced draught for the Taylor stokers has not been deducted from the total steam generated by the boiler. The amount of steam used by the Roney stokers was about 1½ per cent. of the steam generated by the boiler, and for the Taylor stokers about 2½ to 3 per cent.

The 80·98 per cent. efficiency was obtained with the Roney stokers (Test No. 16) with an equivalent evaporation from and at 212° F. of 80,502 pounds, or 3·4 pounds per square foot of heating surface per hour, while the 76·18 per cent. efficiency (Test No. 11A) was obtained with an equivalent evaporation of 175,348 pounds per hour or 7·41 pounds per square foot of heating surface per hour.

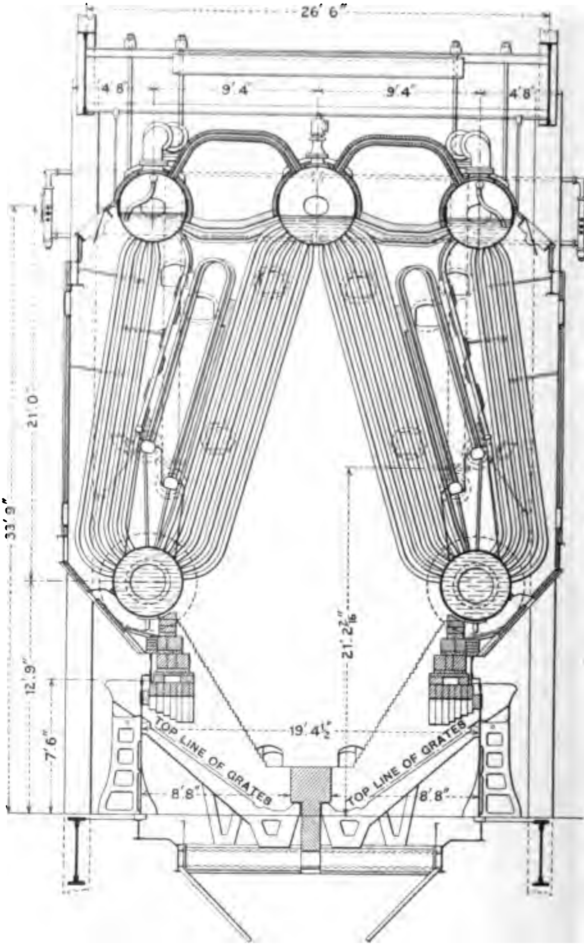


Fig. 140.—Large water-tube boiler installed by the Detroit Edison Co. (End View).

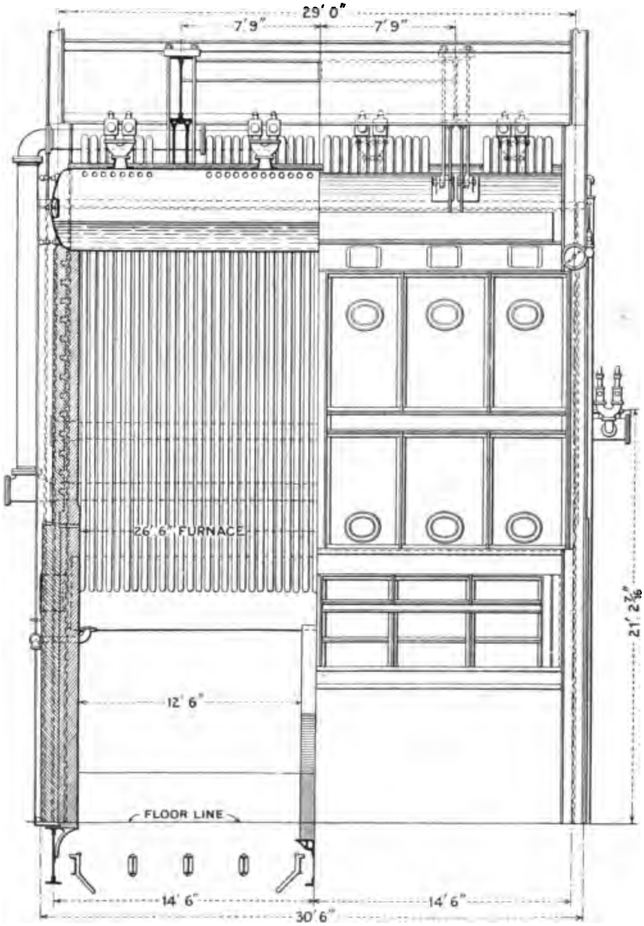


Fig. 140a.—Large water-tube boiler installed by the Detroit Edison Co. (Side Elevation).

TABLE IX.—GENERAL DATA OF TESTS ON LARGE BOILERS OF DETROIT EDISON CO.

With Roney Stokers.

Test No.	Duration (hours).	Steam pressure lbs. per sq. inch (gauge). In boiler.	Entering superheater.	Leaving superheater.	Barometer (inches of mercury).	Average draught inches of water. Air blast in tuyeres.	Suction below boiler dampers.	Suction in ashpit.	Feed temperature ° F.	Temperature of steam ° F. Entering superheater ° F.	Leaving superheater ° F.	Superheat ° F.
1	25	195	193·5	190	29·26	0·42	0·32	0·16	184	383·7	515	135·3
2	24	192	190·2	188	29·17	0·05	0·16	0·06	180	382·8	497	115·2
3	24	198	196·1	192	29·23	0·51	0·39	0·21	184	384·5	516	130·5
4	30	201	199·0	192	29·39	0·55	0·22	0·02	181	384·6	520	136·4
5	24	199	--	187	29·45	0·16	0·24	0·10	183	382·5	491	107·5
6	24	200	197·8	191	29·37	0·57	0·26	0·16	185	384·2	515	136·8
16	32	193	191·4	188	29·37	0·17	0·23	0·12	177	382·9	482	102·1
17	16·5	207	205·5	194	29·39	0·95	0·34	0·06	179	385·4	521	132·6
18	24	207	205·2	194	29·48	1·11	0·33	0·05	178	385·4	540	157·6
2–4 *	90	199	195·7	191	29·27	0·38	0·28	0·06	182	384·1	514	130·9
5–6 *	55	209	195·9	189	29·40	0·38	0·25	0·05	183	383·3	503	125·7

With Taylor Stokers.

7	24	205	202·6	193	29·45	1·30	0·58	0·03	187	384·9	535	155·1
8	24	200	197·0	192	29·46	0·76	0·20	0·11	181	384·6	505	128·4
9	50	206	202·7	192	29·31	1·73	0·53	0·06	184	384·6	538	156·4
10A	24	197	195·0	191	29·50	0·65	0·20	0·15	188	384·2	513	132·8
10	48	195	193·8	190	29·50	0·67	0·20	0·15	187	383·7	515	134·3
11A	12	210	206·5	188	29·25	2·54	0·83	0·03	186	382·8	546	166·2
11	26·5	210	207·5	190	29·25	2·53	0·84	0·02	188	383·7	544	165·3
12	48	197	195·2	189	29·13	0·88	0·26	0·05	179	383·2	519	136·8
14	24	206	203·0	189	29·30	1·56	0·84	0·20	177	383·3	544	167·8
15	24	199	197·3	188	29·28	1·61	0·57	—	176	382·8	524	145·2
7–9 *	109	203	200·6	192	29·30	—	0·45	—	184	384·6	523	147·4
10–11*	80·5	200	198·6	190	29·66	—	0·44	—	187	383·8	525	144·2

* Includes periods between tests.

TABLE X.—GENERAL DATA AND RESULTS OF TESTS OF LARGE BOILERS OF DETROIT EDISON CO.

With Roney Stokers.

Test No.	Duration (hours).	Dry coal fired lbs. per hour.	Equivalent evaporation from and at 212° F. lbs. per hour.	Horse-power developed.	Per cent. of rated H.P. developed on basis of 10 sq. ft. of heating surface per horsepower.	Equivalent evaporation from and at 212° F. per lb. of dry coal.	Equivalent water per hour from and at 212° F. per sq. ft. of heating surface.	Calorific value of coal B.Th.U. per lb.	Temperature of flue gases leaving boiler °F.	Analysis of flue gases leaving boiler (per cent.). CO_2	O_2	CO	Combined efficiency of boiler and furnace (per cent.)	Steam used by stoker engines and steam jets (per cent.).
1	25	7,463	85,948	2491	105·0	11·52	3·63	14,362	576	11·95	7·55	0·05	77·84	—
2	24	5,609	65,671	1903	80·0	11·71	2·78	14,225	480	14·33	4·54	0·11	79·88	—
3	24	8,130	92,840	2691	113·8	11·42	3·92	14,308	542	13·05	6·46	0·18	77·45	0·63
4	30	11,582	124,410	3606	152·4	10·74	5·26	13,756	670	14·74	3·96	0·54	75·78	1·58
5	24	6,606	76,768	2225	94·0	11·62	3·24	13,896	483	14·40	4·54	0·35	81·15	1·75
6	24	11,295	122,984	3565	150·7	10·89	5·20	14,037	662	14·66	4·23	0·31	75·28	1·45
16	32	6,665	80,502	2333	98·6	12·08	3·40	14,476	460	13·55	5·92	0·07	80·98	1·34
17	16·5	13,761	157,722	4572	193·3	11·46	6·67	14,493	636	14·69	4·55	0·20	76·73	1·39
18	24	14,987	159,747	4630	195·7	10·66	6·75	13,689	694	14·16	5·04	0·16	75·57	1·32
2–4 *	90	8,834	97,745	2833	119·8	11·06	4·13	14,098	572	—	—	—	76·13	—
5–6 *	55	9,459	103,905	3012	127·3	10·98	4·39	13,977	575	—	—	—	76·23	—

With Taylor Stokers.

Test No.	Duration (hours).	Dry coal fired lbs. per hour.	Equivalent evaporation from and at 212° F. lbs. per hour.	Horse-power developed.	Per cent. of rated H.P. developed on basis of 10 sq. ft. of heating surface per horsepower.	Equivalent evaporation from and at 212° F. per lb. of dry coal.	Equivalent water per hour from and at 212° F. per sq. ft. of heating surface.	Calorific value of coal B.Th.U. per lb.	Temperature of flue gases leaving boiler °F.	CO_2	O_2	CO	Combined efficiency of boiler and furnace (per cent.)	Steam used by stoker engines and steam jets (per cent.).
7	24	11,094	123,407	3577	151·2	11·12	5·22	14,000	575	14·00	5·50	0·42	77·07	2·61 †
8	24	7,622	88,061	2553	107·9	11·55	3·72	13,965	493	13·69	5·82	0·10	80·28	2·44 †
9	50	11,838	132,935	3853	162·8	11·23	5·62	13,998	574	14·74	4·57	0·12	77·85	2·87 †
10A	24	6,736	76,804	2226	94·4	11·40	3·25	—	489	—	—	—	77·63	—
10	48	6,656	75,808	2197	92·9	11·39	3·22	14,188	487	11·86	7·96	0·06	77·90	2·63 †
11A	12	15,904	175,348	5083	214·8	11·03	7·41	—	668	—	—	—	76·18	—
11	26·5	15,696	172,456	4999	211·3	10·99	7·29	14,061	651	15·45	3·86	0·17	75·84	3·41 †
12	48	8,653	99,032	2870	121·3	11·44	4·18	14,010	535	13·79	5·73	0·04	79·24	2·57 †
14	24	13,469	151,447	4390	185·5	11·24	6·40	14,272	647	14·20	5·08	0·08	76·42	2·95 †
15	24	9,158	100,453	2912	123·1	10·97	4·23	14,213	561	10·83	8·93	0·09	74·90	2·77 †
7–9 *	109	10,209	114,235	3312	140·0	11·19	4·83	13,983	545	—	—	—	77·66	2·68 †
10–11*	80·5	9,868	108,467	3143	132·8	10·99	4·58	14,095	542	—	—	—	75·66	3·04 †

* Includes periods between tests. † Engines driving stokers and steam-turbine driving fans; no steam jets.

TABLE XI.—HEAT BALANCES, DETROIT BOILER TESTS. FLUE GAS ANALYSES AND TEMPERATURE TAKEN IN BREECHING.

WITH RONEY STOKER.

Test No.	Absorbed by boiler.	Moisture in coal.	Hydrogen in coal.	Heat to chimney.			CO.	Carbon in ash.	Radiation, etc.
				Heat to chimney.	Moisture in air.	Total.			
1	77·84	0·18	4·55	13·56	0·37	13·93	0·23	1·20	2·07
2	79·88	0·16	4·36	9·15	0·26	9·41	0·42	1·20	4·57
3	77·45	0·15	4·48	11·39	0·32	11·71	0·74	1·85	3·62
4	75·78	0·17	4·49	12·79	0·36	13·15	1·95	2·13	2·33
5	81·15	0·16	4·29	9·11	0·20	9·31	1·29	2·29	1·51
6	75·28	0·21	4·74	13·17	0·36	13·53	1·16	1·71	3·37
16	80·98	0·22	4·18	8·94	0·19	9·13	0·27	3·20	2·02
17	76·73	0·22	4·47	12·47	0·28	12·75	0·74	2·05	3·04
18	75·57	0·21	4·64	14·34	0·27	14·61	0·62	2·42	1·93
								Average	2·72

WITH TAYLOR STOKER.

Test No.	Absorbed by boiler.	Moisture in coal.	Hydrogen in coal.	Heat to chimney.	Moisture in air.	Total.	CO.	Carbon in ash.	Radiation, etc.
7	77·07	0·18	4·58	11·54	0·28	11·82	1·61	2·31	2·43
8	80·28	0·17	4·28	10·12	0·24	10·36	0·40	1·80	2·71
9	77·85	0·20	4·31	11·21	0·25	11·46	0·44	2·20	3·54
10	77·90	0·18	4·34	11·35	0·26	11·61	0·27	2·77	2·93
11	75·84	0·18	4·47	11·91	0·35	12·26	0·59	3·58	3·08
12	79·24	0·18	4·46	11·05	0·21	11·26	0·16	2·32	2·38
14	76·42	0·18	4·51	13·15	0·27	13·42	0·31	2·57	2·59
								Average	2·81

14. Heat Balances in Locomotive Boilers.—The working conditions with a locomotive boiler running on the road differ considerably from those of a stationary boiler owing to the large amount of fuel which escapes unburnt when a locomotive boiler is forced. As Mr. L. H. Fry * has pointed out, "To complete the heat balance by the usual method of calculation it is necessary to measure accurately the amount of fuel thrown out of the chimney unconsumed; which is a task of very considerable

* See papers in *Proc. Inst. Mech. Eng.*, 1908; in *Engineering* for June 30, 1911, from which we quote; and by Prof. Dalby in *Engineering* for August 26, 1910.

difficulty. Even if the greater part of the combustible in the sparks be measured there must remain uncertainty as to the escape of finely divided carbon such as soot, and of unconsumed volatile matter."

The analysis of the flue gases only enables the weight of gases produced per pound of fuel *actually burned* to be calculated as explained in Art. 7, Chap. II. When the usual method of establishing a heat balance is applied to a locomotive boiler (or, indeed, to any boiler working with a strong mechanically produced draught) there are two unknown quantities introduced, namely, the quantity of heat actually produced (see also Art. 18, Chap. V.), and the weight of the flue gases produced per pound of fuel. Hence the heat lost in the flue gases is unknown until the relation between the weight of coal fired and the weight actually burned is established. This relation can be determined in the following manner:—

Let H_1 = percentage of the heating value of the coal which is lost due to the presence of CO in the smoke-box gases.

H_2 = percentage of the heating value of the coal which is carried away by the flue gases.

E = percentage of the heating value of the coal absorbed by the water plus that lost by radiation.

P = percentage of heating value of the coal lost by unburnt fuel (required).

Then $100 - H_1$ per cent. of the total heating value of the coal is available in the fire-box, and H_2 per cent. escapes into the smoke-box, hence the proportion taken up by the boiler is $100 - H_1 - H_2$ per cent.

The actual heat obtained by combustion in the fire-box is $100 - P$ per cent.

Hence $$E = (100 - P)\frac{100 - H_1 - H_2}{100}$$

or $$P = 100 - \frac{100E}{100 - H_1 - H} \quad . \quad . \quad . \quad . \quad (1)$$

Suppose that from observations made in an actual trial it is

found that 50 per cent. of the heat in the fuel is absorbed by the boiler, of which 45 per cent. is transferred to the water and 5 per cent. lost by radiation, 1·2 per cent. is lost through the formation of CO, and 21·1 per cent. is carried away by the smoke-box gases, then from equation (1) we find

$$\begin{aligned} P &= 100 - \frac{100 \times 50}{100 - 1{\cdot}2 - 21{\cdot}1} \\ &= 100 - 64{\cdot}4 \\ &= 35{\cdot}6 \text{ per cent.} \end{aligned}$$

Hence 35·6 per cent. of the fuel escapes unburnt.

Now the heat carried away in the flue gases is 21·1 per cent. of the heat in the coal burned, or

$$21{\cdot}1 \times 0{\cdot}644 = 13{\cdot}6 \text{ per cent.}$$

of the heat in the coal supplied to the fire-box.

Also, the heat lost through the formation of CO is

$$1{\cdot}2 \times 0{\cdot}644 = 0{\cdot}77 \text{ per cent.}$$

of the heat in the coal fired.

The heat balance, as usually drawn up, will therefore be—

	B.Th.U.	Per cent.
Total heat value of 1 lb. of dried fuel . . .	—	100·0
Heat transferred to water	—	45·00
Heat carried away by the flue gases	—	13·60
Heat lost by incomplete combustion	—	0·77
Heat lost by unburnt fuel	—	35·60
Heat lost by radiation	—	5·0
Unaccounted for	—	0·03
	Total . .	100·00

15. Actual Efficiency of a Boiler.—Although the recommendations of the Institution of Civil Engineers, previously given, have been generally accepted by engineers and form a convenient means of securing uniformity in the tabulation of the results of boiler trials, it may be well to point out that the efficiency calculated by their method is not the true efficiency

of the boiler, inasmuch as no allowance is made for the moisture in the air supply. Exception might also be taken to the method of calculating the efficiency on the calorific value of dried fuel instead of on the calorific value of the fuel as fired.*

The heat in one pound of fuel which is available for steam raising is not that in one pound of dry fuel unless the fuel is perfectly dry when it is used (and this is very rarely the case in practice), because some of the heat of combustion is used up in evaporating and superheating the moisture in the fuel. It would appear more reasonable therefore to place on the credit side of the heat balance the actual available heat in one pound of fuel, *i.e.* the heat in the quantity of dry fuel contained in one pound of the actual fuel as fired, *minus* the heat lost in evaporating and superheating the moisture in one pound of the fuel as fired. In other words, the heat lost in heating, evaporating and superheating to the flue gas temperature the moisture in the fuel should not appear on the debit side of the heat balance. The effect of transferring this item from the debit to the credit side of the heat balance is to raise the efficiency slightly, and although for practical purposes the difference may be considered negligibly small this hardly by itself furnishes sufficient reason for adopting a method which is not strictly correct in principle.

The effect of a moist air supply may also have a very great influence on boiler performance, and it seems strange that this item should have been omitted by the Committee of the Institution of Civil Engineers. On a wet or foggy day a considerable amount of moisture is carried into the boiler furnace for every pound of fuel burned, and this moisture has to be heated, evaporated and superheated, the heat so expended being lost in the flue gases. As in the case of the moisture in the fuel, it hardly seems justifiable to credit the boiler with a heat supply equal to the calorific value of one pound of dried fuel, when the actual amount of heat available will be less than this by the amount of heat expended in heating, evaporating and superheating the moisture in the air supply, as well as in the fuel.

* See an article by Mr. T. B. Morley on "Available Heat in Steam Boilers" (*Engineering*, June 23, 1911), where the above objections are emphasised.

The following example will illustrate the difference between the two methods of calculation and the effect on the efficiency.

Example.—The following data is obtained from a boiler trial:—

Calorific value of dried coal, 13,500 B.Th.U. per pound.

Moisture in 1 pound of coal as fired, 0·10 pound.

Heat transferred to steam per pound of fuel as fired, 8505 B.Th.U.

Temperature of air in boiler house, 75° F.

Temperature of flue gases leaving the boiler, 600° F.

Moisture admitted with the air supply per pound of fuel fired, 0·3 pound.

Heat transferred to steam per pound of dry fuel $= \frac{8505}{0 \cdot 9}$ $= 9450$ B.Th.U.

Heat lost in heating, evaporating and superheating 0·1 pound of moisture to 600° F. will be

$$\{(212 - 75) + 966 + 0{\cdot}5(600 - 212)\} \times 0{\cdot}1$$
$$= \{137 + 966 + 194\} \times 0{\cdot}1$$
$$= 1297 \times 0{\cdot}1 = 129{\cdot}7 \text{ B.Th.U., say } 130 \text{ B.Th.U.}$$

or per pound of dried fuel $= \frac{130}{0 \cdot 9} = 166$ B.Th.U.

According to the recommendations of the Institution of Civil Engineers these two items of the heat account will be:—

	B.Th.U.	Per cent.
Total heat units in 1 lb. of dried fuel	13,500	100·0
Heat transmitted to the water (*thermal efficiency*)	9450	70·0
Heat expended in evaporating and superheating moisture mixed with fuel	166	1·22

Neglecting the moist air supply, these items by the more correct method will be:—

	B.Th.U.	Per cent.
Heat units in 1 lb. of fuel as fired (= heat units of dry coal in 1 lb. of coal as fired—heat absorbed by moisture) $= 0{\cdot}9 \times 13{,}500 - 130 =$	12,020	100·0
Heat transmitted to water (per lb. of fuel as fired)	8505	70·75

Taking into account the moisture in the air supplied as well as the moisture in the fuel, we find :—

Total moisture per pound of fuel as fired $= 0{\cdot}1 + 0{\cdot}3 = 0{\cdot}4$ pound

$\therefore$ Total moisture per pound of dry fuel $= \dfrac{0{\cdot}4}{0{\cdot}9} = 0{\cdot}44$ pound

and heat lost in heating, evaporating and superheating this moisture will be

$$0{\cdot}44 \times 1297$$
$$= 570 \text{ B.Th.U.}$$

By the more correct method we obtain :—

	B.Th.U.	Per cent.
Heat units in 1 lb. of fuel as fired (= heat units of dry coal in 1 lb. of coal as fired—heat absorbed by 0·44 lb. of moisture) $= 0{\cdot}9 \times 13{,}500 - 570$ . . . . $=$	11,580	100·0
Heat transmitted to water (per lb. of fuel as fired)	8505	73·44

From the above it will be seen that in this particular case the actual efficiency of the boiler is 73·44 per cent., or 3·44 per cent. higher than that obtained when the heat balance is drawn up on the total heat in one pound of dry fuel. Neglecting the moisture in the air supply, the efficiency will be 70·75 as against 70 per cent.

The only argument in favour of placing the heat absorbed in evaporating and superheating the moisture in the fuel and in the air supply on the debit side of the heat account is the

simplicity obtained by always reckoning the efficiency on the same quantity, namely, on the heat contained in one pound of dried fuel. The extra complication involved above is however but small, and not nearly as much as that involved in using the Rankine instead of the Carnot cycle for the ideal efficiency of a steam engine, or in taking account of the variability in the specific heat of the gases in a gas engine cylinder when estimating the ideal efficiency of the gas engine.

Examples on Chapter XII.

1. The following particulars were obtained in a test with a throttling calorimeter: Pressure of steam in main steam pipe, 140·3 pounds per square inch gauge; barometer, 29·92 inches; pressure of steam in calorimeter after throttling manometer reading), 2·6 inches; temperature of steam after throttling, 258° F. Estimate the dryness fraction of the steam in the main steam pipe (use steam table, p. 406).

Ans. 0·974.

2. The following data were obtained during a boiler trial:—

Feed water per hour . . .	10,115 pounds.
Temperature of feed to boiler	174° F.
Steam pressure	169·5 pounds per sq. inch absolute.
Moisture in steam per pound	0·019 pound.
Coal fired per hour	1074 pounds.
Dried coal ,, ,, ...	1054 pounds.
Calorific value of dried coal .	14,000 B.Th.U. per pound.
Analysis of dried coal, carbon	88 per cent
hydrogen	3·6 ,, ,,
oxygen	4·8 ,, ,,
ash	3·6 ,, ,,

Calorific value of ashes 900 B.Th.U. per pound.
Weight of ashes per hour 38 pounds.

Analysis of flue gases by volume, CO_2 = 10·9 per cent.
CO = 1·0 ,, ,,
O_2 = 7·1 ,, ,,
N_3 = 81·0 ,, ,,

Temperature of flue gases leaving boiler 600° F.
Temperature of air 60° F.

Draw up a heat balance for this boiler.

Ans.

	B.Th.U.	Per cent.
Total heat value of 1 lb. of dried fuel	14,000	100·0
Heat transferred to water (thermal efficiency) .	9945	71·03
Heat carried away by products of combustion .	1623	11·60
Heat carried away by excess air	861	6·15
Heat lost in evaporating and in superheating moisture mixed with fuel	24	0·17
Heat lost in incomplete combustion	737	5·26
Heat lost by unburnt carbon in ash	32	0·25
Balance of heat account: errors of observation, etc.	778	5·54
	14,000	100·00

3. If the boiler in Question 2 be fitted with a superheater, the flue gases passing from the boiler into the superheater and then on to the economiser, draw up a heat balance for the superheater from the following data :—

Weight of steam entering superheater per hour 10,115 pounds.
Steam pressure 169·5 pounds per sq inch absolute.
Moisture of steam into per pound 0·019 pound.

Temperature of steam into . .	368° F.
,, ,, out of .	400° F.
,, ,, flue gases into .	600° F.
,, ,, ,, out of	500° F.

Mean specific heat of gases leaving superheater 0·24.

Ans.

	B.Th.U.	Per cent.
Heat received from flue gases (per lb. of dried fuel)	2484	100·0
Heat transferred to steam (efficiency of superheater)	303	12·19
Heat carried off in chimney gases	2015	81·12
Balance of heat account	166	6·69
	2484	100·00

PROPERTIES OF SATURATED STEAM.

Absolute press., lbs. per sq. in. p	Temp. ° F. t	Heat of the liquid h	Latent heat of evaporation L	Total heat H	Specific volume, cub. ft. per lb.	Density lbs. per cub. ft.
1	102	70	1035	1105	333	0·0030
2	126	94	1021	1115	173·6	0·00576
3	141·6	109	1012·5	1121·5	118·6	0·00845
4	153	121	1005·6	1126·6	90·4	0·0110
5	162·4	130	1000·0	1130	73·3	0·0136
6	170	138	996	1134	61·9	0·0162
7	177	145	991·7	1136·7	53·5	0·0187
8	183	151	988	1139·0	47·2	0·0212
9	188·4	156·4	984·8	1141·2	42·3	0·0236
10	193	161·0	982	1143·0	38·3	0·0262
14·7	212	180·0	970	1150·0	26·8	0·0374
15	213	181·1	969·7	1150·8	26·3	0·0380
20	228	196·2	960	1156·2	20·0	0·050
25	240	208·5	952	1160·5	16·2	0·062
30	250·3	219·0	944·7	1163·7	13·7	0·073
35	259·2	228·0	938·5	1166·5	11·9	0·084
40	267·2	236·2	933·0	1169·2	10·5	0·095
45	274·4	243·5	928·0	1171·5	9·39	0·106
50	281	250·2	923·3	1173·5	8·50	0 1175
55	287	256·5	918·7	1175·2	7·78	0·1285
60	292·7	262·2	914·6	1176·8	7·18	0·139
65	298	267·7	910·7	1178·4	6·65	0·150
70	303	272·7	907·0	1179·7	6·20	0·161
75	307·6	277·6	903·4	1181·0	5·80	0·172
80	312·1	282·1	900·0	1182·2	5·48	0·183
85	316·3	286·5	896·8	1183·3	5·16	0·194
90	320·3	290·6	893·7	1184·3	4·89	0·204
95	324·1	294·5	890·7	1185·2	4·63	0·216
100	327·8	298·4	887·8	1186·2	4·43	0·226
105	331·4	302·0	885·0	1187·0	4·23	0·236
110	334·8	305·6	882·3	1187·9	4·05	0·247
115	338·1	309·0	879·6	1188·6	3·87	0·258
120	341·3	312·3	877	1189·3	3·72	0·269
125	344·4	315·5	874·6	1190·1	3·58	0·280
130	347·3	318·6	872·2	1190·8	3·44	0·291

PROPERTIES OF SATURATED STEAM.—*continued.*

Absolute press., lbs. per sq in. p	Temp. ° F. t	Heat of the liquid h	Latent heat of evaporation L	Total heat H	Specific volume, cub. ft. per lb.	Density lbs. per cub. ft.
135	350·2	321·6	870	1191·6	3·33	0·301
140	353	324·5	867·5	1192·0	3·21	0·312
145	355·8	327·3	865·3	1192·6	3·11	0·322
150	358·5	330·1	863·1	1193·2	3·01	0·332
155	361·0	332·8	861	1193·8	2·92	0·342
160	363·6	335·4	858·8	1194·2	2·83	0·353
165	366	338	856·8	1194·8	2·75	0·364
170	368·5	340·5	854·8	1195·3	2·68	0·373
175	370·8	343	852·8	1195·8	2·61	0·384
180	373·1	345·5	850·9	1196·4	2·53	0·395
185	375·4	348	849	1197	2·47	0·405
190	377·6	350·2	847·1	1197·3	2·40	0·416
195	379·8	352·5	845·2	1197·7	2·34	0·427
200	381·9	354·6	843·4	1198·0	2·29	0·437
205	384	356·8	841·6	1198·4	2·23	0·448
210	386	359	839·8	1198·8	2·19	0·457
215	388	361	838·1	1199·1	2·14	0·467
220	390	363·1	836·4	1199·5	2·09	0·479
225	391·9	365·1	834·7	1199·8	2·04	0·490
230	393·8	367·5	832·6	1200·1	2·00	0·500
235	395·6	369	831·4	1200·4	1·96	0·510
240	397·5	371	829·8	1200·8	1·92	0·520
245	399·2	372·9	828·2	1201·1	1·89	0·529
250	401	375	826·5	1201·5	1·85	0·541
255	402·7	376·8	825	1201·8	1·81	0·552
260	404·5	378·5	823·5	1202·0	1·78	0·56
265	406·2	380·2	822	1202·2	1·75	0·571
270	407·9	382	820·5	1202·5	1·71	0·585
275	409·5	383·8	819	1202·8	1·68	0·595
280	411·2	385·5	817·5	1203·0	1·66	0·60
285	412·7	387·2	816·1	1203·3	1·63	0·615
290	414·4	388·8	814·7	1203·5	1·60	0·625
295	415·9	390	813·7	1203·7	1·58	0·634
300	417·5	392	812·0	1204·0	1·55	0·645

INDEX

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